Energy Research and Development Division
FINAL PROJECT REPORT

RADIANT HEATING AND COOLING AND MEASURED HOME PERFORMANCE FOR CALIFORNIA HOMES

Prepared for: California Energy Commission
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PREFACE

The California Energy Commission Energy Research and Development Division supports public interest energy research and development that will help improve the quality of life in California by bringing environmentally safe, affordable, and reliable energy services and products to the marketplace.

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Radiant Heating and Cooling and Measured Home Performance for California Homes is the final report for the Advanced Radiant HVAC Systems for California Homes project (contract number 500-08-051) conducted by Gas Technology Institute. The information from this project contributes to Energy Research and Development Division's Building's End-Use Energy Efficiency Program.

For more information about the Energy Research and Development Division, please visit the Energy Commission's website at www.energy.ca.gov/research/ or contact the Energy Commission at 916-327-1551.

ABSTRACT

This report summarized the findings of a project demonstrating the feasibility of low cost residential radiant heating and cooling systems coupled with measured home performance techniques to meet the space conditioning load. The radiant cooling system design featured off-peak chilled water generation and storage for load shifting and the radiant heating system used a high-efficiency natural gas as a combined hot water source for space and domestic water heating. Other key design components included radiant surfaces, piping and manifolds, pumps and valves and electronic controls. Results of modeling the thermal performance of the radiant panels and storage tank were presented. Measured home performance was used to reduce the load of the two field test sites, including reducing the load on the heating, ventilation and air conditioning system through upgrading the envelope, distribution system and other key components while simultaneously measuring the effectiveness of the upgrades. Measured home performance refers to techniques that create a more comfortable and safer home with measurably higher energy savings.

Two Northern California field test sites were upgraded with measured home performance techniques and then retrofitted with the radiant systems. Utility billing data showed a 45 percent average reduction in heating energy for the two sites, adjusted for weather. Data for cooling savings clearly showed both a significant peak load reduction (95 percent) and an overall power use reduction (19 percent). Additional analysis supported better temperature stability than with a forced air system, having a fast response to thermostat settings in the absence of humidity and condensation issues.

Barriers to adoption of the proposed system were considered from the point of view of customers, manufacturers, contractors and utilities.

Keywords: California Energy Commission, comfort, consumer savings, cooling, demand reduction, heating, high-efficiency, hydronic systems, incentives, load shifting, market barriers, measured home performance, off-peak, radiant, radiant panels, residential, retrofit, thermal storage

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TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

EXECUTIVE SUMMARY

Introduction

There are more than 7.9 million single family detached housing units in California (U.S. Census 2011), 97 percent of which are equipped with a space conditioning system (EIA RECS 2013). Over half of the 2012 new single family housing units were built in hot dry climates where real estate costs were lower than in cooler coastal areas (U.S Census 2012). Compressor-based cooling systems are routinely being installed in these applications. Over 60 percent of residential central heating, ventilation and air conditioning (HVAC) systems in California are forced air (EIA RECS 2013). Rated air conditioner efficiency can be as high as 18 Seasonal Energy Efficiency Ratio (SEER), but the effective efficiency when considering the entire system—the house--is still relatively low. Factors such as fan power, duct leakage, improper refrigerant charge and lower efficiency during hot outdoor conditions can reduce the net efficiency significantly compared to rated efficiency. In addition, the lower efficiency at high outdoor temperatures coincides with peak load, which results in the need for additional generation capacity with poor load factor. Residential air conditioning's seven percent statewide load factor has adverse impacts on electric utilities.

Residential buildings in California generally have hard ceiling surfaces and slab on grade or crawl space construction for the floors. Floors were typically carpeted but hardwood floors were not uncommon. Hydronic radiant floor heating systems, where used, are placed under hard floor surfaces or within slabs, and cooling is seldom a consideration. Radiant heating is generally regarded as a luxury feature due to high pricing and a reputation for excellent comfort. A Davis Energy Group PIER project completed in 2005 (Berman, 2005) showed how radiant floor heating costs could be reduced to be attractive for production homes, but there was no evidence of any uptick in that market. Ceilings offered more promise for radiant cooling when compared to radiant floors from the perspective of both performance (carpets and floor coverings can reduce the cooling capacity and trap moisture) and comfort (cool floors may be uncomfortable to walk on). Past installations have been mainly in Europe, although some commercial buildings in the United States are designed with cooling panels that fit into "T-bar" suspended ceilings. These systems are expensive to install ($10 per square foot) but perform well.

Condensation on the surface of cooled panels is an issue that restricts cooling rates in humid climates. Typical room cooling loads can be fully satisfied in the dry western United States without danger of condensation from 55 degree Fahrenheit (°F) cooling water (Stetiu, 1999).

Radiant systems have some advantages over forced air systems: no filter maintenance, less distribution of airborne contaminants in the home and reduced noise. Many home and hotel occupants complain about forced air noise from registers and wall air conditioning (AC) units, as confirmed in a 1998 Davis Energy Group study (Davis, 1998).

The advantages of radiant heating systems are well-known to HVAC engineers. Chapter 6 of the 2008 *ASHRAE Handbook of HVAC Systems and Equipment* provided both technical information and references for a variety of designs, including floor and ceiling mounted options for residences. Radiant heating comfort benefits were well-documented (see for example Chapter 8 of the 2005 *ASHRAE Handbook of Fundamentals*). Other benefits included high energy efficiency, no noise, low-temperature water and unobtrusive location. Hydronic radiant systems were strongly preferred over electric radiant systems due to installed cost and operating efficiency advantages. Hydronic radiant systems were seldom used in residential construction, however.

Project Purpose

The overall goal of this project was to demonstrate the efficacy of hydronic radiating systems when coupled with integrated installation techniques and to promote increased adoption in California new residential construction so that related electric peak-shifting benefits could be captured. The secondary goal of this project was to use hardware testing in the lab and in the field to determine application barriers and critical success factors for cost-effective implementation of this system with integrated installation.

The project objectives were to:

1. Evaluate design options and establish the cost effectiveness of a low-cost, ceiling mounted, residential radiant cooling system with chilled water storage for peak-shifting.
2. Evaluate design options for the accompanying hydronic heating system.
3. Study the use of integrated installation techniques using measured home performance to reduce the space conditioning load and the corresponding radiant system cost and to improve the comfort in the space.

Project Results

Radiant heating and cooling systems were coupled with measured home performance techniques to reduce the overall space conditioning load and to improve space comfort from reduced air infiltration. Measured home performance differed from the traditional fragmented industries providing design and installation practices for insulation, air sealing and HVAC systems. These three functions were typically performed by different subcontractors in both new construction and retrofit markets. The results were acceptable from a cost perspective because they provided a low first-cost expense. The bidders who could comply with the primary measurement criteria of lowest cost and quickest installation that meets code requirements succeeded. The results were extremely poor from an energy perspective. Current codes do not require a measured energy performance, notwithstanding the qualitative methods provided by Title 24. The consumer therefore has few means of objectively judging the value of air sealing, insulation and HVAC systems other than their separate installed costs. The energy savings were substantial when these elements were measured and integrated, and the installed cost could be equal to the often poorly-performing, non-integrated current practices.

Three ceiling-mounted radiant panels were evaluated: (1) a manifold design with straight tubes; (2) a serpentine design with a single curved tube; and (3) a commercial panel supplied by Uponor. Laboratory testing of the non-commercial panels showed that the serpentine design provided better performance at 11 British thermal units per hour per square foot (Btu/hr/ft^2) at the cooling design condition and 16 16 Btu/hr/ft^2 at the heating design condition. The Uponor panel was part of their European product line. The panel tested at 16 Btu/hr/ft^2 for cooling and 33 Btu/hr/ft^2 for heating, matching the manufacturer's literature. The smaller commercial panels had shorter embedded tube lengths for better heat transfer.

Two prototype tank designs were developed, a hard tank and a soft tank. The hard tank was intended for use outside the house, with a capacity of approximately 570 gallons and R-31 foam insulated walls. The soft tank was a four-feet high and six-feet diameter cylindrical outer shell made of high tensile fabric, an inner liner and four inches of foam between the two liners. Laboratory testing for both tanks showed that the heat gain under test conditions at an ambient temperature of 87°F was less than 400 Btu per hour. Buoyant convection cooled the tank adequately without the need for additional mechanical circulation of the water across the coil during laboratory cooling tests. The hard tank was chosen for the field tests because the field test sites did not have basements.

Field tests were carried out at two locations in Sacramento, California: Grandstaff Drive and 6th Avenue. Both systems were installed as retrofits in single-story single family homes of approximately 1000 square feet. The 6th Avenue house used the serpentine tube panels designed in the project, with individual panels plumbed between the supply and return manifolds. The Grandstaff Drive house used the Uponor panels, with a central manifold and 18 circuits of up to six panels each. This allowed for better control of flow balance than the 6th Avenue layout. In both cases the panels were attached directly to the existing ceiling and interconnections were made in the attic. Monitoring was carried out continuously through a complete cooling and heating season.

The results of the cooling system field tests supported the findings in the laboratory. The capacity of the storage tanks was never exhausted during peak hours, reducing peak cooling energy use by 95 percent. The shifting of the compressor use to night time allowed further savings, calculated at 19 percent, due to a higher coefficient of performance (COP) at lower nighttime ambient temperatures when compared to a non-storage system. Spot ventilation in the bathrooms and kitchens was adequate to remove moisture generated by the occupants, so the system was never shut down by the dew point control. Temperature stability was better with the radiant system than the forced air system, and this was achieved without sacrificing recovery from setback, which was measured at an initial 3°F per hour.

The high efficiency water heater and radiant panels met the load at both houses during the winter for space heating. Gas energy savings from Grandstaff was 34 percent compared to the baseline from a utility bill analysis. The thermostat setpoint was lowered from 70°F to 68°F in the first month of testing in this house by the homeowner due to the improved thermal environment. Energy savings from natural gas would have been slightly lower if the setting would have been left at 70°F for the entire heating season. Energy savings from natural gas was 57 percent when compared to the baseline in the 6th Avenue house. The energy reduction associated with the radiant heating system and the measured home performance improvements produced an average savings of 45 percent for the two houses in the Sacramento area. It was not possible to separate the effects of the two factors in this study, however a predicted savings for the increase in efficiency of the heating plant alone would yield a 15 percent savings for

Grandstaff (vs. 80 percent for an Annual Fuel Utilization Efficiency [AFUE] furnace) and 30 percent savings for 6th Avenue (vs. 65 percent for an efficient heating system). This left approximately 25 percent of the savings to be spread between thermal envelope improvements, the performance of the radiant heating system and the use of a lower thermostat setpoint.

Economic analysis of the cost of traditional HVAC systems and the radiant system design showed an incremental cost of approximately $3000 in a mature market when installed in new construction and factoring in the savings from the elimination of ductwork. An estimated energy saving for cooling of approximately 20 percent of baseline electric consumption yielded a payback of between five years and 15 years without peak-shifting incentives from the utility in California climate zones 10 and 12. This estimate was supported by the field test results. Adding incremental heating system efficiency improvements could reduce energy costs from 10 – 45 percent depending on the starting point and could also improve the payback for the system. Using integrated installation techniques with measured home performance significantly reduced the energy consumption of the house since less chilled water storage was required.

Training in measured home performance and radiant heating and cooling systems was conducted by Pacific Gas and Electric (PG&E), Sempra, and Southern California Edison (SCE) to accomplish technology transfer. Approximately 150 students learned about the latest techniques to provide quality installation while simultaneously measuring the effectiveness of the upgrade. Students also received a briefing on this project. Feedback from attendees was overwhelmingly positive.

Technology transfer for this project took several other forms to reach a variety of audiences:

1. A softcover book, *Measured Home Performance, Guide to Best Practices for Home Energy Retrofits in California* by Rick Chitwood and Lew Harriman is available on Amazon.com.
2. A consumer's guide to measured home performance.
3. A contractor's guide to measured home performance.
4. Eighteen short videos featuring measured home performance techniques.
5. Four training classes.
6. A website: http://measuredhomeperformance.com/ containing the videos, a link to the softcover book for free download and the consumer's guide and contractors guides.
7. Technical papers in several forums.

Project Benefits

This project demonstrated the feasibility of low cost residential radiant heating and cooling systems coupled with measured home performance techniques to meet the space conditioning load. Radiant heating and cooling systems can reduce the energy used for both heating and cooling. Using less energy helps reduce greenhouse gas emissions that contribute to climate change and also reduces other emissions that cause air pollution.

CHAPTER 1: Introduction

There are more than 7.9 million single family detached housing units in California (U.S. Census 2011), 97 percent of which are equipped with a space conditioning system (EIA RECS 2013). Over half of the 2012 new single family housing units built in California were located in hot dry climates, where real estate costs are lower, rather than in cooler coastal areas (U.S Census 2012). Compressor-based cooling systems are routinely being installed in these applications. Over 60 percent of residential central HVAC systems in California are forced air (EIA RECS 2013). While rated air conditioner efficiency can be as high as 18 SEER, the effective efficiency when considering the entire system—the house--is still relatively low. Factors such as fan power, duct leakage, improper refrigerant charge, and lower efficiency at hot outdoor conditions can reduce the net efficiency significantly compared to rated efficiency. In addition, the lower efficiency at high outdoor temperatures coincides with peak load, resulting in the need for additional generation capacity with poor load factor. Residential air conditioning's seven percent statewide load factor has adverse impacts on electric utilities.

Residential buildings in California generally have hard ceiling surfaces and slab on grade or crawl space construction for the floors. Floors are typically carpeted but hardwood floors are not uncommon. Hydronic radiant floor heating systems, where used, are placed under hard floor surfaces or within slabs; cooling is seldom a consideration. Radiant heating is generally regarded as a luxury feature due to high pricing and a reputation for excellent comfort. A Davis Energy Group PIER project completed in 2005 (Berman, 2005) showed how radiant floor heating costs could be reduced to be attractive for production homes, but there is no evidence of any uptick in that market. When compared to radiant floors, ceilings offer more promise for radiant cooling from the perspective of both performance (carpets and floor coverings can reduce the cooling capacity and trap moisture) and comfort (cool floors may be uncomfortable to walk on).

Condensation on the surface of cooled panels is an issue that restricts cooling rates in humid climates. In the dry western U.S., typical room cooling loads can be fully satisfied without danger of condensation from 55°F cooling water (Stetiu, 1999). Previous work by Davis Energy Group has also confirmed this observation (ORNL 1998). For example, the All-Weather NightSky system in Vacaville is now in its 11th cooling season.

Since carpets interfere with thermal delivery, radiant ceilings and walls typically offer greater cooling savings than floors. Though past installations have been mainly in Europe, many commercial buildings in the Chicago area have used radiant ceiling cooling for decades, with panels that fit into "T-bar" suspended ceilings. These systems are expensive to install (about $10 per square foot) but perform well. Recent research reports (Stetiu, 1999 and Conroy and Mumma, 2001) show the U.S. potential for radiant ceiling cooling. But T-bar ceilings are not compatible with typical residential construction, where drywall ceilings predominate. Pre-fabricated radiant surfaces are also available which are lightweight and respond rapidly to a change in inlet water temperature.

Radiant systems have some advantages over forced air systems: no filter maintenance, less distribution of airborne contaminants in the home (Mohamed, 2010), and reduced noise. Many home and hotel occupants complain about forced air noise from registers and wall AC units, as confirmed in a 1998 Davis Energy Group study (Davis, 1998).

In this project, radiant heating and cooling systems were coupled with measured home performance techniques to reduce the overall space conditioning load and improve comfort from reduced air infiltration. Measured home performance as practiced by a few specialists in California differs from the traditional method where fragmented industries providing design and installation practices for insulation, air sealing and HVAC systems. Standard practice in both new construction and retrofit markets, these three functions are performed by different subcontractors who neither communicate with each other nor measure the results. From a cost perspective, the results are acceptable, because they provide a low first-cost solution. The bidders who can comply with the primary measurement criteria of lowest cost and quickest installation that meets code requirements succeed.

From an energy perspective, the results are extremely poor. Current codes do not require measured energy performance. So the consumer has no means to objectively judge the value of air sealing, insulation and HVAC systems other than their separate installed costs. The small-but-critical differences in design and installation which improve energy performance are not rewarded by commercial success. However, when these elements are measured and integrated, the energy savings are substantial and the installed cost can be equal to the poorly-performing, non-integrated current practices.

For example, at the same site in Redding, CA, two identical houses were erected in the same development of high-end, energy-efficient homes. Installed costs were similar for both homes. The first house used conventional design and installation practices, but used advanced technology heating and cooling equipment (a geothermal heat pump). The second only used conventional heating and cooling equipment, but the home used integrated design and installation with measured home performance techniques. That home was able to maintain comfort using a system with less than 30 percent of the cooling capacity of non-integrated equipment sizing assumptions. Its measured annual energy consumption was 60 percent less than the home which used non-integrated design and installation, in spite of the theoretical energy advantage of that home’s geothermal heat pump (Springer, 2006).

In this project, Gas Technology Institute (GTI) managed a comprehensive program that integrated radiant cooling, heating, and related envelope systems and installation methods in new and existing California homes. The linked projects are designed to reduce system costs while significantly increasing the installed efficiency of residential space conditioning systems in cooling-dominated climates throughout California, especially hot dry climates.

A research team comprising 15 organizations, including two not-for profit research organizations, seven small businesses (including a DVBE), two utilities, three manufacturers, and one design/build firm collaborated on this project. In addition to these team members, noncontractual participants included two major building product manufacturers and two additional utilities who have worked successfully with the team members in past programs.

The program targets results that benefit both ongoing mandatory (Title 24) and voluntary (utility energy efficiency program) efforts to achieve greater energy efficiency in residential space conditioning systems. The program also provided valuable insights for manufacturers and installers ranging from component selection to installation best practices to field performance for emerging advanced energy efficiency design and installation options. Together, this will result in substantial energy and cost savings to California consumers, while helping to reduce peak demand for electric power used for space conditioning during extreme temperature periods.

This project was designed around the following Energy Commission target areas:

1. HVAC systems (target area 2)

   The work allows the development of an entirely new class of residential cooling systems for new construction and retrofit markets in hot, dry climate zones.

   The practical guidance and data produced throughout the course of this program are market-focused, giving this approach considerable market credibility; making it possible to implement the technology quickly through utility energy efficiency programs.

2. Building Envelope (target area 1)

   Advanced integrated installation methods are developed that provided significant improvements to the as-installed performance of the building envelope as well as the HVAC system. (Chitwood, 2011; Chitwood, 2012)

   High efficiency envelope installation methods are developed for new construction and retrofit applications that will significantly reduce the required size and peak power consumption of the HVAC system, accelerating the transition to carbon neutral homes.

1.1 Objectives

The overall objective of this project is to demonstrate the efficacy of radiant heating and cooling systems when coupled with integrated installation techniques that address envelope deficiencies. A secondary objective is to provide test results that will support increased adoption of these systems in California housing and capture the electric peak-shifting benefits of radiant cooling.

The advantages of radiant heating delivery are well-known to HVAC engineers. Chapter 6 of the 2008 *ASHRAE Handbook of HVAC Systems and Equipment* provides both technical information and references for a variety of designs, including floor and ceiling mounted options for residences. Radiant heating comfort benefits are well-documented (see for example Chapter 8 of the 2005 *ASHRAE Handbook of Fundamentals*) and include high energy efficiency, no noise, low-temperature water for heating, improved comfort, and unobtrusive equipment in the space. In this project, these advantages are applied to residential radiant heating and cooling systems that employ ceiling-mounted hydronic panels, chilled water storage for peak-shifting, and a high-efficiency combination water and space heating system to reduce cost and increase efficiency.

This project is broken into several smaller projects:

1. Radiant cooling systems
2. Radiant heating systems
3. Integrated installation with measured home performance
4. Technology transfer.

The detailed program objectives follow the project breakdown. They are:

1. Radiant Cooling
    a. Evaluate the potential for residential radiant cooling in California
    b. Develop and lab-test preferred products
    c. Field test developed products
    d. Identify preferred market paths
    e. Evaluate and report project results
2. Radiant heating
    a. Evaluate integrated and supplemental radiant heating design options
    b. Lab-test preferred products and methods
    c. Field test developed products in conjunction with cooling products
3. Integrated installation with measured home performance
    a. Provide guidance on current best practices for integrated installation methods
    b. Develop advanced integrated installation methods for emerging technologies
    c. Demonstrate guidance in conjunction with heating and cooling tests
4. Technology transfer
    a. Develop classroom, online, and consumer-grade materials based on project results
    b. Conduct classroom training sessions at participating utility sites
    c. Host a multimedia website containing subject expert videos and other materials
    d. Provide consumer-grade materials to participating utilities in a suitable format

CHAPTER 2: Radiant Cooling

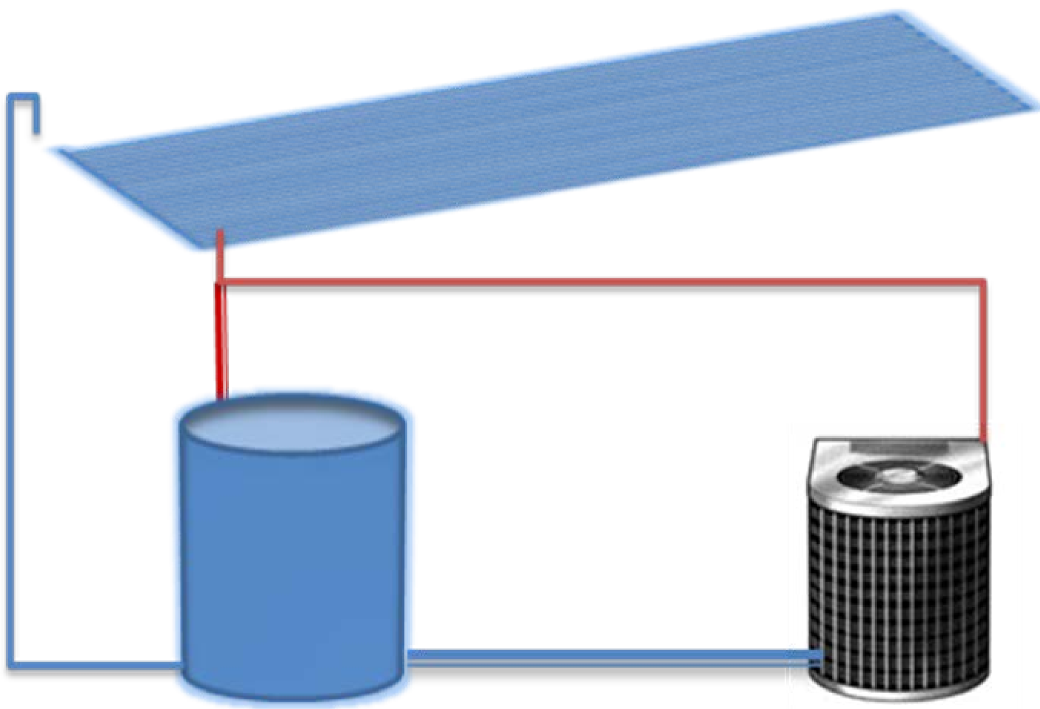
2.1 Purpose

The primary purpose of the radiant cooling part of the project is to design and implement an energy-saving off-peak radiant cooling system for single-family homes. The ideal market for the technology is hot and dry climate zones where dehumidification is not required. This report shows the feasibility of using radiant cooling technology across hot and dry California climate zones through evaluating the economic potential of the technology, developing and testing prototype equipment in the lab, conducting field tests, and evaluating the energy savings.

2.2 System Design

The radiant cooling system has three major components: radiant surface, chiller, and thermal storage. Figure 1 shows the layout of the system.

Figure 1: Panel, Chiller, and Storage Tank Schematic



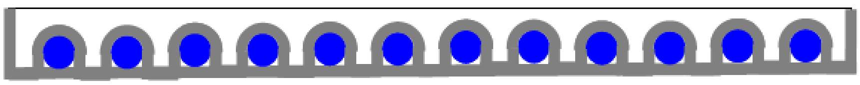
Source: Western Cooling Efficiency Center

2.2.1 Radiant Surface

The radiant element presents a cool surface to the room which absorbs thermal radiation by using chilled water flowing through an array of tubes. The team has considered arrays both above and below the drywall ceiling. The former strategy is less expensive as it does not require a finished surface but less versatile because it requires attic space above. For the economic evaluations, the team has focused on the more expensive approach of securing rigid panels below the ceiling. The team evaluated market acceptability of two designs, with both potentially using compressed glass fiber board as the structural element and having designs which require no *wet* finishing to complete the installation.

The first would rout grooves for "pressed-in" tubes. The lower face of the panel will then be covered with an aluminum foil that acts as a surface and heat exchange fin. Figure 2 shows this design in schematic cross-section.

Figure 2: Cross-Sectional View of Routed Panel



Source: Western Cooling Efficiency Center

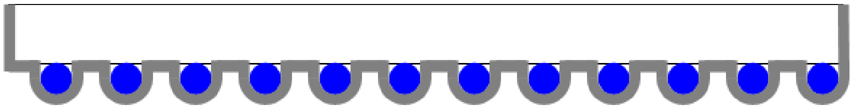
From below, the routed design appears as a flat panel as shown in Figure 3. The surface would be finished as a standard ceiling.

Figure 3: Rendering of Flat Radiant Panels

Photo Credit: Western Cooling Efficiency Center

The second is a ribbed design, Figure 4, which requires no routing for the parallel circuit tubes, instead expressing them as ribs, on the panel surface. The larger surface area of aluminum foil would improve performance compared to the routed panel, and the ribs could potentially help hide panel joints.

Figure 4: Cross-Sectional View of Ribbed Panel



Source: Western Cooling Efficiency Center

Figure 5 shows a "looking upward" rendering of the ribbed design. Again, panels would be off-white, but paintable to match any room color.

Figure 5: Rendering Of Ribbed Radiant Panels

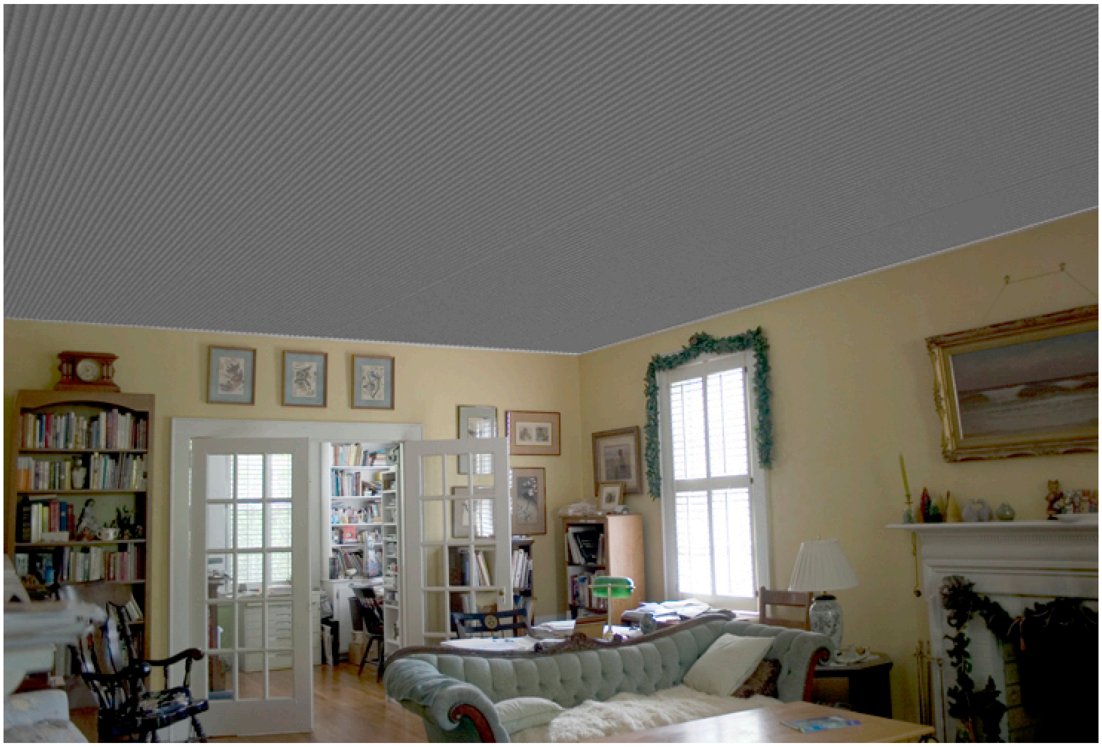
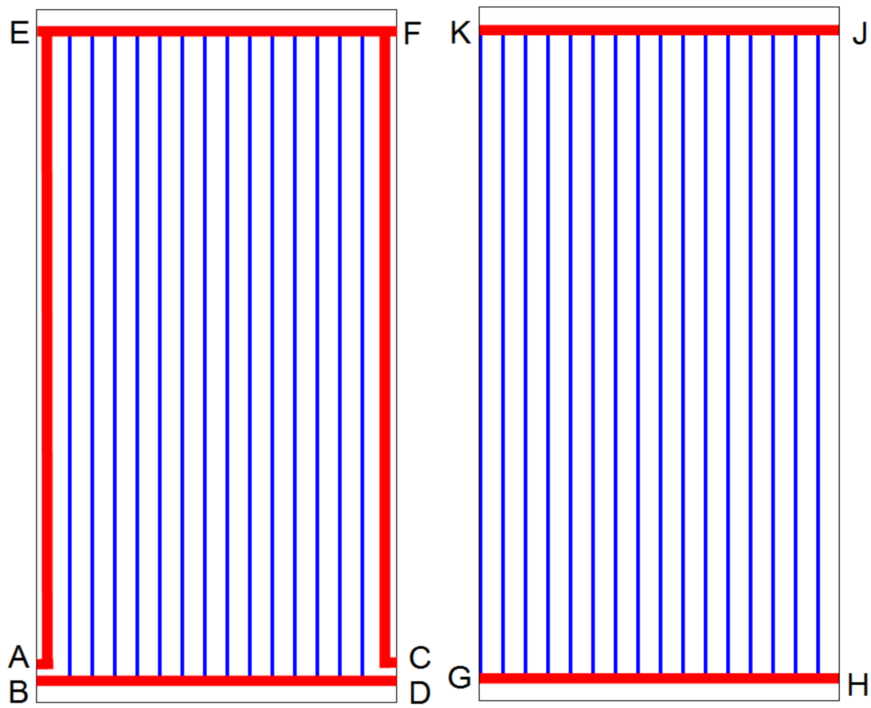


Photo Credit: Western Cooling Efficiency Center

With either the routed or the ribbed design, only two panel types are needed, as shown schematically in Figure 6. The manifolds are the thick red lines and the cooling tubes are the thinner blue lines. The first layout (left) would be used when a single panel is needed. Water inlet and outlet are at A and B respectively. All other tube ends (C through F) would be sealed. For most rooms, one or more additional simple panels (Figure 6) would be placed next to the first panel. Outlets D and G are connected together, as are F and K. Outlets C, E, H and J remain sealed. The layout of the first panel is symmetric to allow the attachment of additional panels on either side.

During the prototyping phase of the project it became clear that the cost and complexity of routing the required grooves into the panel to create the flat panel was beyond the scope of this project, so the ribbed design was chosen for further development.

Figure 6: Panel Arrangement Tubing Layout

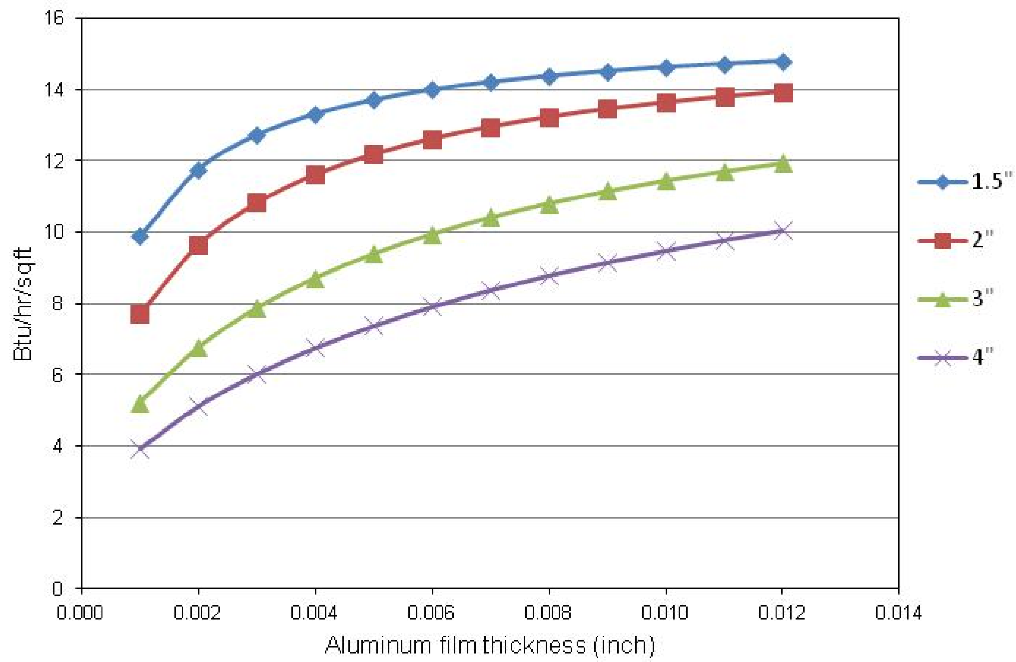


Source: Western Cooling Efficiency Center

2.2.2 Panel Thermal Analysis

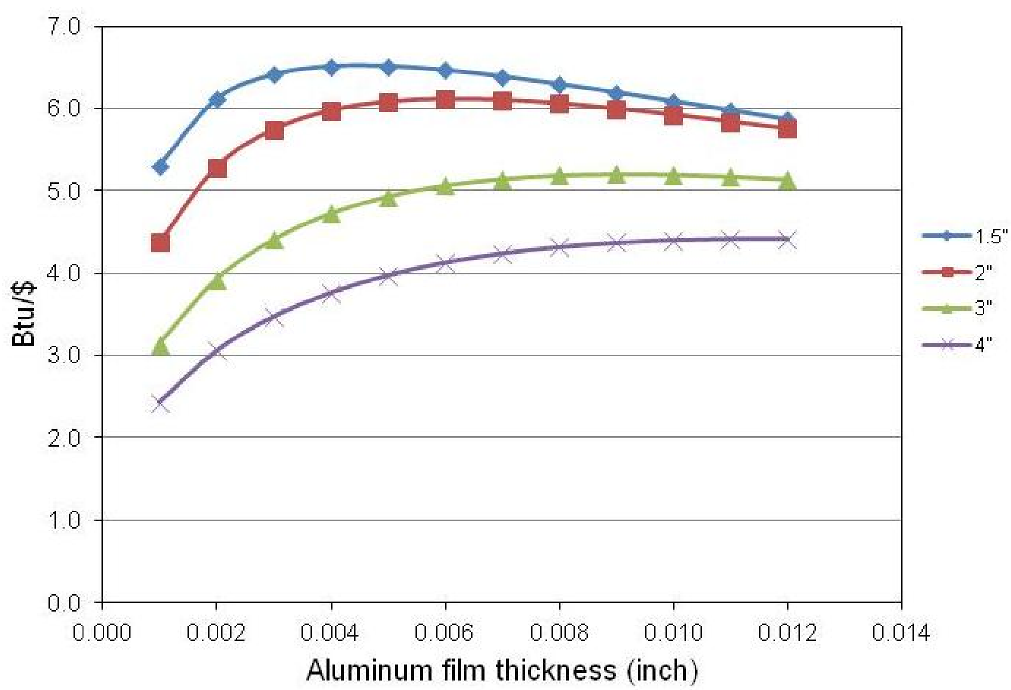
The team modeled thermal performance to verify anticipated cooling delivery. The method used is based on a model by Conroy and Mumma (Conroy 2001). The graphs below summarize the pertinent results. All the calculations are done with a room Average Uncontrolled Surface Temperature (AUST) of 76°F and a panel temperature of 58°F. The numbers given for Btu per square foot are purely radiative and do not include a convective component.

Figure 7: Performance of Radiant Panels as a Function of Film Thickness for Various Tube Spacing



Source: Western Cooling Efficiency Center

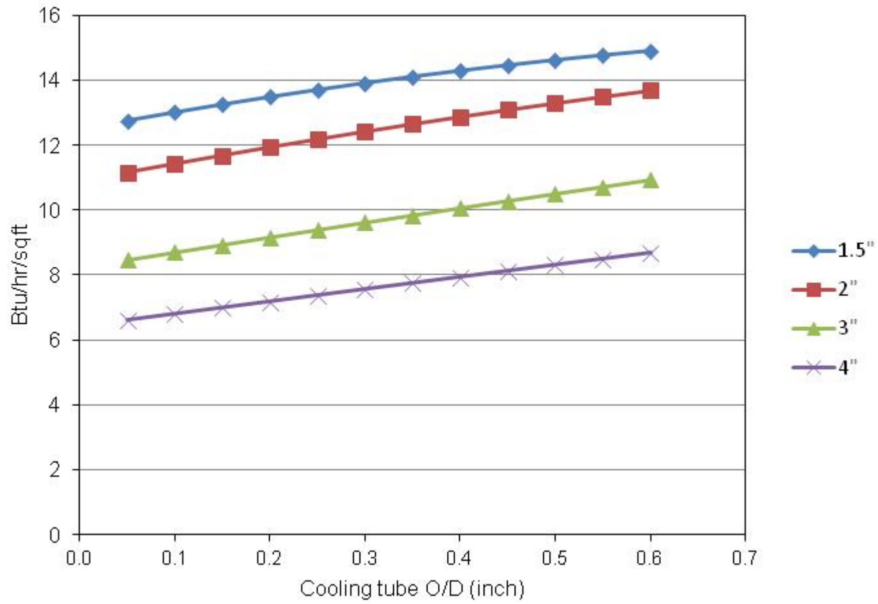
Figure 8: Btu per Dollar as a Function of Film Thickness for Various Tube Spacing



Source: Western Cooling Efficiency Center

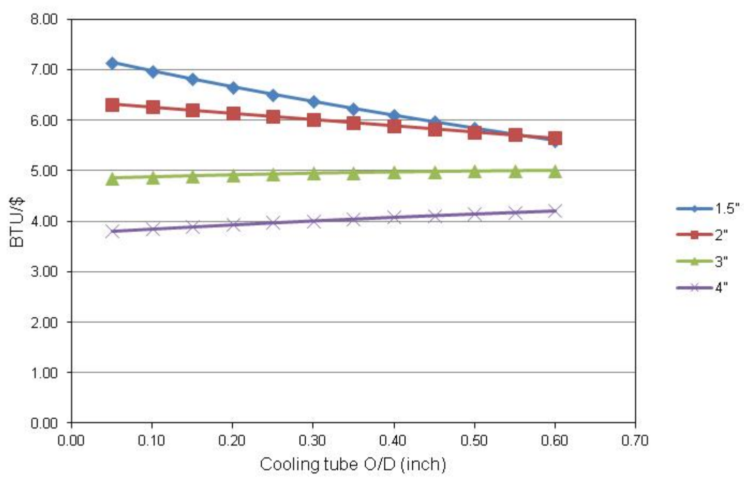
The optimum film thickness, in terms of the balance of first cost and performance, is approximately five thousandths of an inch. This is the value used for the next calculation, of the effect of tube diameter:

Figure 9: Performance of Radiant Cooling Panels as a Function of Tube Diameter for Various Tube Spacing



Source: Western Cooling Efficiency Center

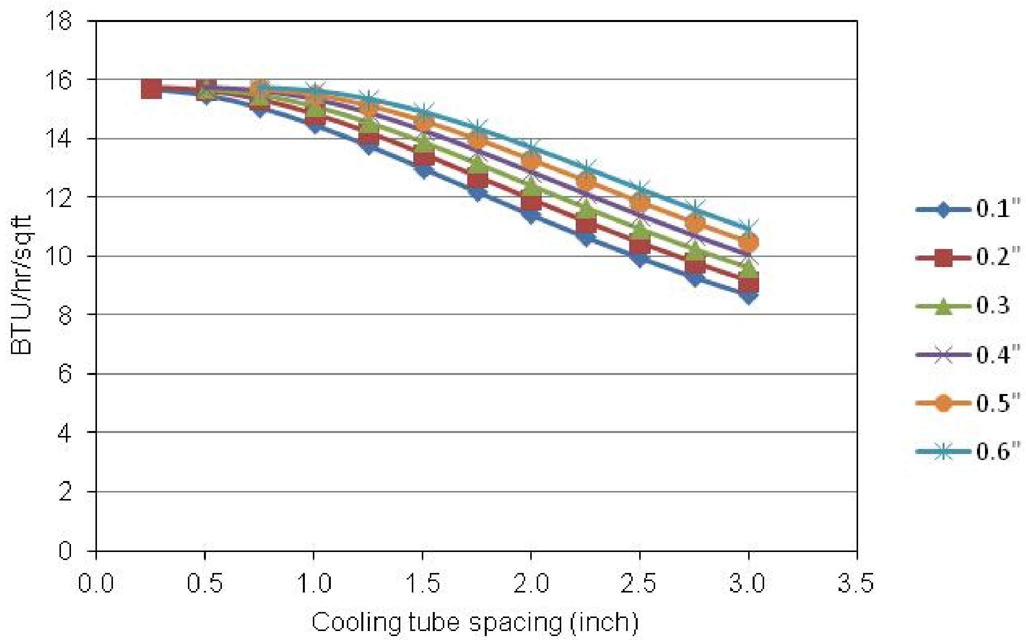
Figure 10: Btu per Dollar as a Function of Tube Diameter for Various Tube Spacing



Source: Western Cooling Efficiency Center

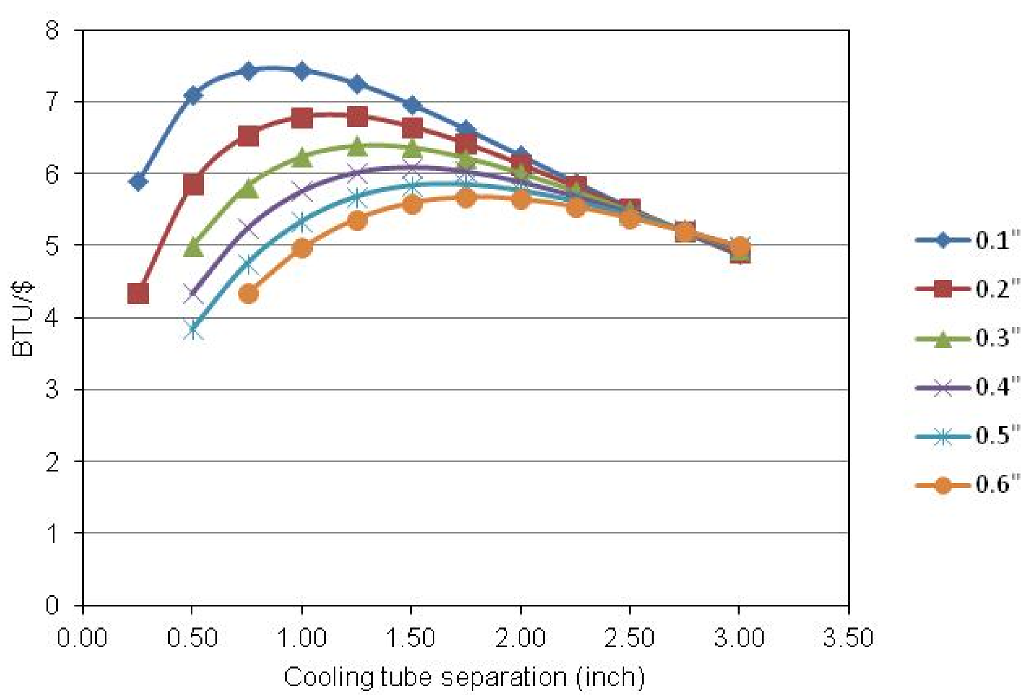
From these graphs it appears that smaller tubes are more cost effective but provide less cooling per unit cost. The following graphs show the effect of tube spacing for a number of different diameters.

Figure 11: Performance of Radiant Panels as a Function of Tube Spacing For Various Tube Diameters



Source: Western Cooling Efficiency Center

Figure 12: Btu per Dollar as a Function of Tube Spacing For Various Tube Diameters



Source: Western Cooling Efficiency Center

From these graphs, one can see that the best compromise between the performance, as determined by the number of Btu per square foot per hour, and the price, as given by the number of Btu per dollar cost, will be approximately:

1. Aluminum film thickness: 0.005 inch
2. Cooling tube diameter: 0.25 inch
3. Cooling tube spacing: 1.5 inch

2.2.3 Storage

A wide range of water and phase-change storage options were considered in the preliminary evaluation stage. The storage capacity will vary by climate zone and house size. On the basis that the system is designed not to be used during peak hours, the storage capacity will need to be—at a minimum—capable of meeting the total cooling requirements for the on-peak hours. These hours are from one p.m. to seven p.m. in all climate zones except for 12, where on-peak hours are 12 p. m. to six p. m. to correspond with the Sacramento Municipal Utility District's (SMUD) time-of-use schedule. Approximate tank sizes can be seen in Table 1. These are based on a 58°F panel inlet water temperature and a 10°F temperature rise across the panels.

Table 1: Required Storage Tank Size by Climate Zone

Climate Zone #	Highest daily peak cooling load (Btu)		Storage capacity required (gal)	
	1 story	2 story	1 story	2 story
2	94494	128296	1200	1800
8	81544	108805	1400	1700
9	96170	138016	1500	1900
10	104520	135458	1700	2100
12	100495	129114	1600	1900
13	129103	162757	1800	2200
15	125969	155066	1800	2200

Source: Western Cooling Efficiency Center

The project considered two storage models. The simplest model is a cylindrical tank which would sit outside the house. In light of the size of tank this would require, the team also modeled a low profile storage tank which could be hidden under a deck. This design was considered in spite of the higher surface to volume ratio, which would increase heat transfer to the surroundings. Renderings of the cylindrical storage tank and deck storage tank can be seen in Figure 13 and Figure 14 below.

Subsequent to this modeling the design of the system was modified to reduce the size of the storage tank required by reducing the water storage temperature to 38°F and including a mix-back loop in the supply to maintain a panel inlet temperature of 58°F. This redesign eliminated the need for the under deck storage tank.

Figure 13: Rendering Of Tank Storage

Photo Credit: Western Cooling Efficiency Center

Figure 14: Rendering Of 'Deck' Storage Option

Photo Credit: Western Cooling Efficiency Center

2.2.4 Energy Consumption

Radiant cooling systems, installed in hot dry climates, consume less energy than traditional forced air units because there is no unnecessary latent heat removal, duct losses, or blower heat being added to the space. These advantages translate into both energy savings and cooling cost savings. Table 2 summarizes simulated yearly energy consumption savings and cooling cost savings for a radiant system over a forced air unit.

Table 2: Forced Air vs. Radiant System Yearly Energy Consumption and Cooling Cost

	Base Case Forced Air with Tier Structure		Radiant System with Time of Use Structure	
Climate Zone & House	Yearly kWh	Yearly Cooling Cost	Yearly kWh	Yearly Cooling Cost
CZ 2 - 1764 ft² 1 story	936	$153	466	$82
CZ 2 - 2312 ft² 2 story	1676	$323	809	$151
CZ 8 - 1764 ft² 1 story	1266	$292	714	$148
CZ 8 - 2312 ft² 2 story	2075	$503	1146	$253
CZ 9 - 1764 ft² 1 story	1828	$394	965	$169
CZ 9 - 2312 ft² 2 story	2674	$606	1395	$255
CZ 10 - 1764 ft² 1 story	2394	$512	1220	$263
CZ 10 - 2312 ft² 2 story	3347	$739	1699	$387
CZ 12 - 1764 ft² 1 story	2069	$242	994	$108
CZ 12 - 2312 ft² 2 story	2923	$363	1394	$151
CZ 13 - 1764 ft² 1 story	3529	$667	1885	$308
CZ 13 - 2312 ft² 2 story	5058	$1,126	2711	$488
CZ 15 - 1764 ft² 1 story	6887	$1,481	3773	$816
CZ 15 - 2312 ft² 2 story	8656	$2,008	4948	$1,171

Source: Western Cooling Efficiency Center

The reduction in energy consumption and the cooling cost savings are displayed in Table 3 below.

Table 3: Yearly Energy Consumption and Cooling Cost Savings for Radiant Systems vs. Forced Air

	Electric Energy Savings, kWh		Cost Savings, $	
Climate Zone & House	kWh	%	$	%
CZ 2 - 1764 ft² 1 story	470	50%	$71	46%
CZ 2 - 2312 ft² 2 story	867	52%	$172	53%
CZ 8 - 1764 ft² 1 story	552	44%	$144	49%
CZ 8 - 2312 ft² 2 story	929	45%	$250	50%
CZ 9 - 1764 ft² 1 story	863	47%	$225	57%
CZ 9 - 2312 ft² 2 story	1279	48%	$351	58%
CZ 10 - 1764 ft² 1 story	1174	49%	$249	49%
CZ 10 - 2312 ft² 2 story	1648	49%	$352	48%
CZ 12 - 1764 ft² 1 story	1075	52%	$134	55%
CZ 12 - 2312 ft² 2 story	1529	52%	$212	58%
CZ 13 - 1764 ft² 1 story	1644	47%	$359	54%
CZ 13 - 2312 ft² 2 story	2347	46%	$638	57%
CZ 15 - 1764 ft² 1 story	3114	45%	$665	45%
CZ 15 - 2312 ft² 2 story	3708	43%	$837	42%

Source: Western Cooling Efficiency Center

2.2.5 Peak Demand Reduction

By switching from the base case forced-air cooling system to the radiant system with on-peak storage, there will be a peak demand reduction. Since the radiant system stores enough energy to provide cooling to the house throughout on-peak hours, no vapor compression will need to occur during these hours. Hence, there will be peak demand reduction seen by the utilities. For the one-story simulation home, Table 4 shows this reduction by climate zone.

Table 4: Single Story 1764 Square Feet Peak Demand Reduction (Undiversified)

Climate Zone	kW reduction
2	3.02
8	2.94
9	3.9
10	4.24
12	4.12
13	4.26
15	5.17

Source: Western Cooling Efficiency Center

In the same manner, for the 2-story simulation home, Table 5 shows peak demand reduction.

Table 5: Two Story 2312 Square Feet Peak Demand Reduction (Undiversified)

Climate Zone	kW reduction
2	4.17
8	3.97
9	4.79
10	5.33
12	4.95
13	5.54
15	6.39

Source: Western Cooling Efficiency Center

2.2.6 Humidity

Unlike traditional forced air cooling systems, a purely radiant cooling system (without a separate dehumidification system) cannot be operated to dehumidify indoor air. The majority of California has a dry climate, but humidity could be a concern in a few areas, namely areas closer to the coast. In these areas, radiant cooling can still be applicable, with the assistance of a small dehumidifier. Table 6 displays the amount of time during the May through October cooling season where indoor simulated humidity levels are within the ASHRAE humidity comfort zone. It appears valley climate zones show the most promise for radiant systems and coastal zones may need additional dehumidification.

Table 6: Humidity Levels

Number of Hours of Year Load Unmet - With and Without Dehumidifier		
Climate Zone & House	Without Dehumidifier	With Dehumidifier
CZ 2 - 1764 ft² 1 story	0	0
CZ 2 - 2312 ft² 2 story	4	0
CZ 8 - 1764 ft² 1 story	44	0
CZ 8 - 2312 ft² 2 story	228	0
CZ 9 - 1764 ft² 1 story	58	0
CZ 9 - 2312 ft² 2 story	192	3
CZ 10 - 1764 ft² 1 story	4	0
CZ 10 - 2312 ft² 2 story	58	1
CZ 12 - 1764 ft² 1 story	5	0
CZ 12 - 2312 ft² 2 story	59	7
CZ 13 - 1764 ft² 1 story	25	0
CZ 13 - 2312 ft² 2 story	175	53
CZ 15 - 1764 ft² 1 story	29	0
CZ 15 - 2312 ft² 2 story	246	17

Source: Western Cooling Efficiency Center

2.2.7 Cost and Payback

Based on the cost models developed for the radiant cooling systems, the team has calculated the payback period for one and two story houses for each climate zone. The storage in each case is assumed to be the tank type shown in Table 7. The payback period is given for three different scenarios: The first case assumes no incentive from the utility companies. The second case assumes a rebate of $600 for every kW of peak demand removed, using an assumed diversity of 80 percent. The third case assumes a rebate of $1200 for every kW of peak demand removed, again using an assumed diversity of 80 percent. In accordance with the assumptions used in the original proposal, all parties need to see this technology as “distributed generation,” displacing the need to build new generators while reducing energy consumption and combustion of fossil fuels. Several existing California utility programs provide incentives of $1600 per kW and higher, yet these technologies do not save energy and only minimally reduce fossil fuel consumption. Thus, the $1200 per kW incentive appears justified given the significant system advantages.

Table 7: Single Story Costing and Payback

Single story house		Climate Zone 2	8	9	10	12	13	15
1	Inputs							
a	Base case tons	2.5	2.5	3.5	3.5	3.5	4	5
b	Base case cost	$3,450.00	$3,450.00	$3,750.00	$3,750.00	$3,750.00	$3,900.00	$4,200.00
c	Base case $/yr	$153.00	$292.00	$394.00	$512.00	$242.00	$667.00	$1,481.00
d	radiant sqft	1111	1764	1764	1764	1764	1764	1764
e	storage gallons	1400	1500	1600	1800	1700	2000	2000
f	new tons	2	2	2	2	2.5	2.5	4
2	Performance							
a	kW savings	3.02	2.94	3.9	4.24	4.12	4.26	5.17
b	kWh savings	430	494	778	1076	1000	1501	2780
3	Costs							
a	radiant $/sqft	2	2	2	2	2	2	2
b	storage $/gallon	0.81	0.77	0.75	0.7	0.72	0.66	0.66
c	radiant $	$ 2,222.00	$ 3,528.00	$ 3,528.00	$ 3,528.00	$ 3,528.00	$ 3,528.00	$ 3,528.00
d	storage $	$ 1,134.00	$ 1,155.00	$ 1,200.00	$ 1,260.00	$ 1,224.00	$ 1,320.00	$ 1,320.00
e	source $	$ 1,100.00	$ 1,100.00	$ 1,100.00	$ 1,100.00	$ 1,300.00	$ 1,300.00	$ 1,900.00
f	piping &hardware $	$ 400.00	$ 400.00	$ 400.00	$ 400.00	$ 400.00	$ 400.00	$ 400.00
g	Total $	$ 4,856.00	$ 6,183.00	$ 6,228.00	$ 6,288.00	$ 6,452.00	$ 6,548.00	$ 7,148.00
h	Incremental $	$ 1,406.00	$ 2,733.00	$ 2,478.00	$ 2,538.00	$ 2,702.00	$ 2,648.00	$ 2,948.00
4	Economics							
a	$/yr cooling	$ 93.00	$ 163.00	$ 184.00	$ 291.00	$ 116.00	$ 344.00	$ 922.00
b	$/yr saved	$ 60.00	$ 129.00	$ 210.00	$ 221.00	$ 126.00	$ 323.00	$ 559.00
c	Payback/years							
	with no incentive	23.4	21.2	11.8	11.5	21.4	8.2	5.3
	with $600/kW	0.0	10.3	2.9	2.3	5.8	1.8	0.8
	with $1200/kW	0.0	0.0	0.0	0.0	0.0	0.0	0.0
d	Interest rate	0%	0%	0%	0%	0%	0%	0%

Source: Western Cooling Efficiency Center

Table 8: Two Story Costing and Payback

Two story house		Climate Zone 2	8	9	10	12	13	15
1	Inputs							
a	Base case tons	4	4	5	5	5	5	5
b	Base case cost	$3,900.00	$3,900.00	$4,200.00	$4,200.00	$4,200.00	$4,200.00	$4,200.00
c	Base case $/yr	$323.00	$503.00	$606.00	$739.00	$363.00	$1,126.00	$2,008.00
d	radiant sqft	2312	2312	2312	2312	2312	2312	2312
e	storage gallons	2000	1800	2000	2300	2100	2200	2400
f	new tons	2.5	2.5	3.5	3.5	3.5	3.5	4
2	Performance							
a	kW saving	4.17	3.97	4.79	5.33	4.95	5.54	6.39
b	kWh savings	805	840	1166	1516	1411	2347	3322
3	Costs							
a	radiant $/sqft	2	2	2	2	2	2	2
b	storage $/gallon	0.66	0.7	0.66	0.61	0.64	0.62	0.6
c	source $/ton	300	300	300	300	300	300	300
d	radiant $	$ 4,624.00	$ 4,624.00	$ 4,624.00	$ 4,624.00	$ 4,624.00	$ 4,624.00	$ 4,624.00
e	storage $	$ 1,320.00	$ 1,260.00	$ 1,320.00	$ 1,403.00	$ 1,344.00	$ 1,364.00	$ 1,440.00
f	source $	$ 1,300.00	$ 1,300.00	$ 1,700.00	$ 1,700.00	$ 1,700.00	$ 1,700.00	$ 1,900.00
g	piping &hardware $	$ 400.00	$ 400.00	$ 400.00	$ 400.00	$ 400.00	$ 400.00	$ 400.00
h	Total $	$ 7,644.00	$ 7,584.00	$ 8,044.00	$ 8,127.00	$ 8,068.00	$ 8,088.00	$ 8,364.00
i	Incremental $	$ 3,744.00	$ 3,684.00	$ 3,844.00	$ 3,927.00	$ 3,868.00	$ 3,888.00	$ 4,164.00
4	Economics							
a	$/yr cooling	$ 93.00	$ 163.00	$ 184.00	$ 291.00	$ 116.00	$ 344.00	$ 922.00
b	$/yr saved	$ 230.00	$ 340.00	$ 422.00	$ 448.00	$ 247.00	$ 782.00	$ 1,086.00
c	Payback/years							
	with no incentive	16.3	10.8	9.1	8.8	15.7	5.0	3.8
	with $600/kW	7.6	5.3	3.7	3.1	6.0	1.6	1.0
	with $1200/kW	0.0	0.0	0.0	0.0	0.0	0.0	0.0
d	Interest rate	0%	0%	0%	0%	0%	0%	0%

Source: Western Cooling Efficiency Center

2.2.8 Building Envelope

Any improvement in the building envelope which reduces heat gains and air infiltration will also reduce both peak demand and annual consumption. This will be true irrespective of the cooling system used. For a forced air system, the tonnage required will be reduced. For the radiant system with off- peak storage proposed here, there will be a reduction in the area of the radiant panels required, a reduction in the storage capacity needed and a reduction in the chiller tonnage required.

The model house used in the previous simulation was adjusted to have a tighter, better insulated envelope to allow a cost comparison between the forced air and radiant cooling systems as part of a whole house improvement. Table 9 and Table 10 show the tonnage requirements, storage requirements (for the radiant case), and cooling costs for the modified houses in climate zones 10 and 12. Looking at just the forced air system, the cost savings due to the retrofit is between 54 percent and 66 percent in this model, broadly in line with reported savings from Chitwood Energy (Chitwood 2011).

Table 9: Modeled Performance of Forced Air System after Envelope Improvements

Base Case Forced Air System					
Climate Zone	# of Stories	AC size (tons)	Annual usage (kWh)	Peak demand (kW)	Yearly Tiered Cost
10	1	2	1122	2.44	$211 (SDGE)
10	2	2.5	1714	3.14	$341 (SDGE)
12	1	2	803	2.48	$82 (SMUD)
12	2	2.5	1342	3.21	$144 (SMUD)

Source: Western Cooling Efficiency Center

Table 10: Modeled Performance of Radiant System after Envelope Improvements

Radiant System						
Climate Zone	# of Stories	Fraction of ceiling covered	Compressor size (tons)	Tank Size	Annual usage (kWh)	Yearly TOU Cost
10	1	53%	1	800	408	$66 (SDGE)
10	2	54%	1	1200	675	$113 (SDGE)
12	1	56%	1	800	252	$27 (SMUD)
12	2	59%	1	1100	492	$53 (SMUD)

Source: Western Cooling Efficiency Center

The team utilized these results and applied them to the overall cost model shown in Table 7 and Table 8. The new costing and payback periods are shown in Table 11 for single and two story houses respectively. Comparison of these tables with the original versions suggest that while the energy savings for the radiant system are lower with a better building envelope, the economic case is not significantly different, due to the savings made by being able to reduce the panel area and the storage size. Such savings are not carried across to the forced air case.

Table 11: Costing and Payback Period with Envelope Improvements

Single story house		Climate Zone 10	Climate Zone 12	Two story house		Climate Zone 10	Climate Zone 12
1	Inputs			1	Inputs		
a	Base case tons	2	2	a	Base case tons	2.5	2.5
b	Base case cost	$3,300.00	$3,300.00	b	Base case cost	$3,450.00	$3,450.00
c	Base case $/yr	$211.00	$82.00	c	Base case $/yr	$341.00	$144.00
d	radiant sqft	935	988	d	radiant sqft	1249	1364
e	storage gallons	800	800	e	storage gallons	1200	1200
f	new tons	1	1	f	new tons	1	1
2	Performance			2	Performance		
a	kW savings	2.44	2.48	a	kW savings	3.14	3.21
b	kWh savings	714	1000	b	kWh savings	1039	850
3	Costs			3	Costs		
a	radiant $/sqft	2	2	a	radiant $/sqft	2	2
b	storage $/gallon	1.14	1.14	b	storage $/gallon	0.88	0.88
c	radiant $	$1,870.00	$1,976.00	c	radiant $	$2,498.00	$ 2,728.00
d	storage $	$ 912.00	$ 912.00	d	storage $	$1,056.00	$ 1,056.00
e	source $	$ 700.00	$ 700.00	e	source $	$ 700.00	$ 700.00
f	piping &hardware $	$ 400.00	$ 400.00	f	piping &hardware $	$ 400.00	$ 400.00
g	Total $	$3,882.00	$3,988.00	g	Total $	$4,654.00	$ 4,884.00
h	Incremental $	$ 582.00	$ 688.00	h	Incremental $	$1,204.00	$ 1,434.00
4	Economics			4	Economics		
a	$/yr cooling	$ 66.00	$ 27.00	a	$/yr cooling	$ 113.00	$ 53.00
b	$/yr saved	$ 145.00	$ 55.00	b	$/yr saved	$ 228.00	$ 91.00
c	Payback/years			c	Payback/years		
	with no incentive	4.0	12.5		with no incentive	5.3	15.8
	with $600/kW	0.0	0.0		with $600/kW	0.0	0.0
	with $1200/kW	0.0	0.0		with $1200/kW	0.0	0.0
d	Interest rate	0%	0%	d	Interest rate	0%	0%

Source: Western Cooling Efficiency Center

The issue of humidity generation in the house is one that is hard to resolve due to the variety of moisture generating activities that are possible and the variation in activity between households. After consideration of available data, the team felt that it would be realistic to leave the moisture generation level in the model at 10 kg/day for each house, which is in line with ASHRAE standard 162.

Table 12 shows the effect of dehumidification on the original model houses. It is clear from the modeling that substantial reductions in the area of panels required, and therefore in the installed cost of the system, can be made by the use of dehumidification. However, during field testing condensation was only seen in the bathroom of the 6th Avenue house prior to the installation of the extractor fan (see Figure 51) which suggests that humidity levels are overestimated in the model.

Table 12: Modeled Dehumidification Requirements

Coverage Needed to have no hours with load unmet			
Climate Zone	# of Stories	Without Dehumidifier	With Dehumidifier
2	1	42%	34%
2	2	46%	37%
8	1	100% (6 hours unmet)	35%
8	2	100% (8)	36%
9	1	100% (18)	40%
9	2	100% (20)	43%
10	1	97%	40%
10	2	99%	42%
12	1	100% (2)	42%
12	2	100% (2)	43%
13	1	100% (4)	50%
13	2	100% (5)	49%
15	1	100% (6)	51%
15	2	100% (6)	48%

Source: Western Cooling Efficiency Center

2.3 Prototyping

The team investigated and built prototypes of two panel designs. Both consist of cooling tubes situated with an aluminum skin acting as a thermal collector and a fiberboard backing sheet for insulation. The aluminum side faces down and absorbs radiant heat from the room. The aluminum is in direct thermal contact with the cooling tubes and acts as an extended heat transfer surface. The panels are nominally four feet by eight feet, this being the standard size of the fiberboard panels the team utilized. Both prototypes ended up somewhat longer than eight feet to allow for the tubing connections.

The first design has a tubing 'network' with a supply and return header at opposite ends of the panel connecting the cooling tubes which run along the length of the panel. The second design has a single cooling tube running in a serpentine pattern. The tubing layouts are shown schematically in Figure 6.

For the network design, the cooling tubes are on two inch spacing, with a five mil thick aluminum skin. During construction of the prototype it became clear that the aluminum skin was not sufficiently robust to withstand the inevitable impacts that the panels would endure during shipping and installation. For this reason, the prototype of the serpentine design was built using significantly thicker aluminum, 32 mil. The aluminum thus provides structural strength as well as promoting heat transfer. The serpentine design has cooling tubes spaced six inches apart and in both cases the lower face of the aluminum is primed to increase its emissivity.

For the network design, the manifolds are intended to be recessed into the fiberboard, and the panel length will therefore be the same as the length of the fiberboard. For the serpentine design, the aluminum is eight to 10 inches longer than the fiberboard, allowing the bends in the tube, and the interconnects between panels to be hidden above the aluminum. The aluminum used is sufficiently stiff to allow this design to withstand handling and mounting.

The panels are designed to be fitted below the existing plasterboard ceiling and to be installed without the need for a wet finishing process. This allows them to be used in both new build and retrofit applications.

2.3.1 Fabrication

The prototypes for lab testing were built to be functionally equivalent to the intended finished product.

2.3.1.1 Network or manifold design

Figure 15: Prototype of the Network Panel

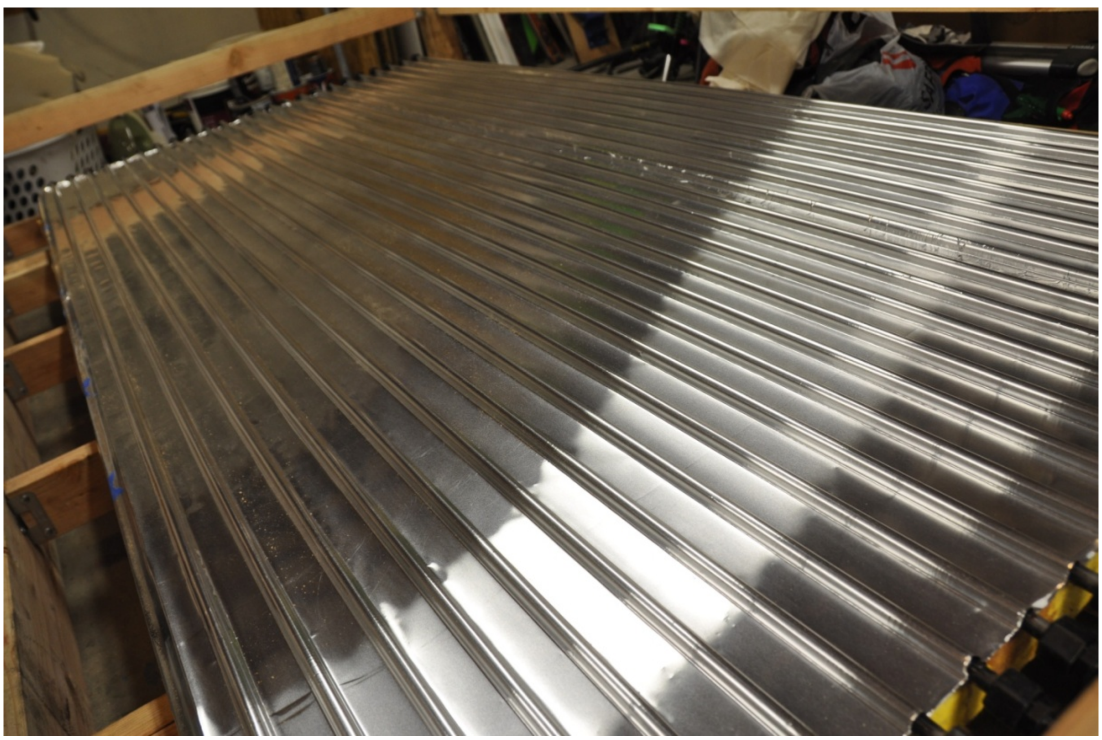


Photo Credit: Western Cooling Efficiency Center

The cost analysis of the network design is based on the proposition that a monolithic tubing network can be fabricated by using extruded tubing for both the manifolds and the cooling tubes, which are then welded together. The low cost of the extruded tubing would allow the network to be made at low cost once appropriate tooling had been made. The cost of tooling required for this design (estimated at greater than $50,000) is too high for the budget of this project (see section on cost analysis). Therefore, the team built the prototype using cooling tubes attached to copper manifolds using “push-fit” fittings. This allows the cooling surface to be functionally identical to that of the intended final design so that the team can test the model without the cost of tooling. The prototype has a plywood stiffener between the fiberboard and the aluminum film, which would not be used in a final product. This was added to stiffen the assembly to cope with the additional mass of the copper manifolds. This plywood also helps hold the cooling tubes in place as the tubing used in this prototype panel was rolled and therefore has a tendency to curl rather than lie flat.

Initial tests showed that attaching the tubing to the fiberboard panels and then rolling the aluminum skin over it would be extremely difficult, due mainly to the near-impossibility of keeping the cooling tubes flat and parallel while unrolling the aluminum and the tendency of the aluminum to buckle and fold while being unrolled. Therefore, the team built a jig from a four foot by eight foot sheet of medium density fiberboard by routing out 17/32 inches wide, 5/8 inches deep grooves on two inch spacing. Using a half inch diameter steel bar, the aluminum was pressed into the grooves. Once each groove was formed, it was coated with thermally conductive grease, and a tube was pressed into it. After all 24 cooling tubes were in place, a quarter inch thick plywood sheet was glued to the surface using spray on adhesive. The two inch fiberboard insulation was glued to the upper side of the plywood. The manifolds were attached to the cooling tubes using "push-fit" connectors. Finally, one inch by four inch planks were attached along the top of the panel and the manifolds were supported by attaching them to the planks.

2.3.1.2 Serpentine tube design

Figure 16: Serpentine Tube Panel

Photo Credit: Western Cooling Efficiency Center

The prototype for the serpentine design was fabricated using a pressed aluminum sheet. The panel is shown with tubing in place prior to attachment of fiberboard backing. The return bends in the tubing are out of the plane of the aluminum, allowing the grooves to continue, giving a uniform appearance from below.

Figure 17: View of the Underside of the Serpentine Tube Panel

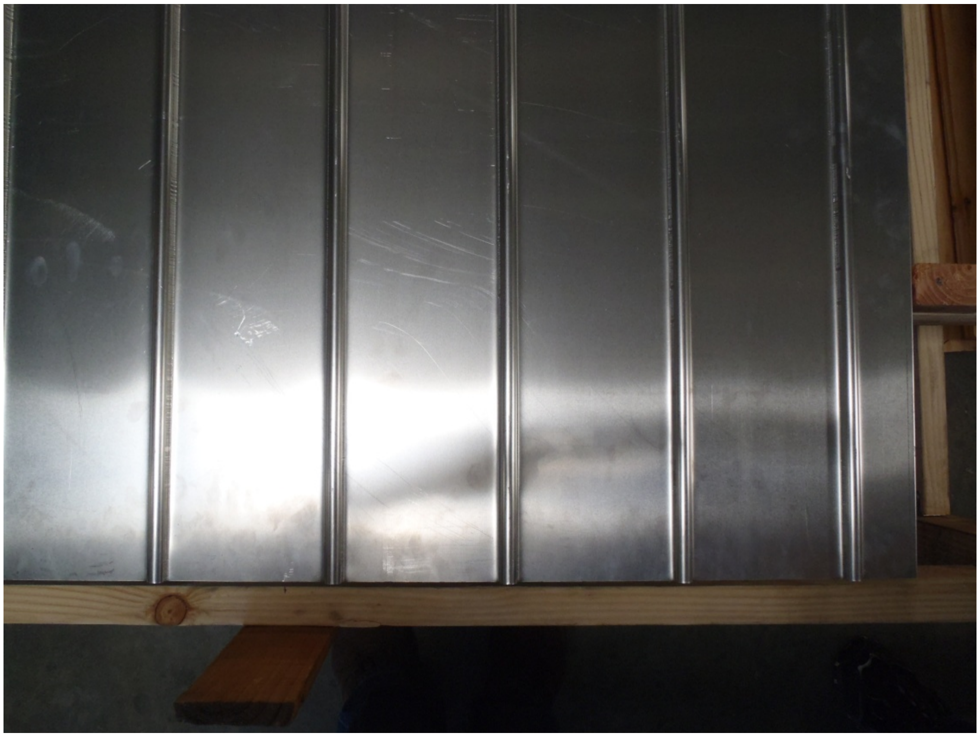


Photo Credit: Western Cooling Efficiency Center

The prototype for this design is close to the anticipated final look. The aluminum was shaped by a local metal working firm. The tube was bent by hand, using a heat gun to soften it. The grooves were coated with thermally conductive grease before the tubing was pushed in. Actual fabrication would use a jig, possibly heated, to ensure uniformity of the tube winding onto the panels. The final version of the panels used in the field test was fitted with a wooden frame, primarily for aesthetic reasons. The frame had the additional benefit of providing secure mounting points to attach the panels to the ceiling.

2.3.2 Prototyping Thermal Analysis

A thermal analysis of the proposed panel designs was carried out as shown in Section 2.2.2, above. The lab testing stage of the project is intended, in part, to verify the conclusions of the models presented in that report. From those models, radiant cooling of ~13Btu/hr/ft^2 (for a water inlet temperature of 58°F and average uncontrolled surface temperature (AUST) of 76°F) for the network design and approximately 11 Btu/hr/ft^2 for the serpentine design is expected. This does not include the additional cooling provided by natural convection. It should be noted here that the thickness of the aluminum skin has a significant impact on the thermal performance and that the cost of bulk aluminum has varied by a factor of three over the course of the project. It is therefore unrealistic to settle on a specific aluminum thickness. From a thermal point of view, the aluminum should be as thick as possible.

2.3.2.1 Conclusions

The thicker aluminum sheet on the serpentine panel makes it more robust than the network design. Thermally, the benefit of the closely spaced tubes in the network design allows the use of thinner aluminum, but this benefit is offset by the increased fragility of the panel compared to the serpentine design. For reasons of ease of manufacture, cost, and robustness, the serpentine design was used in the field tests.

2.3.3 Storage Tank

2.3.3.1 Design

The design criteria for the storage tank are as follows

- Thermal insulation sufficient to ensure that the chilled water does not heat up excessively between overnight chilling and daytime use
- Volume is large enough to provide sufficient stored chilled water to meet the load during peak hours without additional chilling
- Stratification of the chilled water to make maximum use of the stored capacity

The initial system design called for the water to be chilled to the temperature at which it would be circulated through the panels (58°F). This would have required storage capacities of up to 2200 gallons, which was felt to be prohibitively large for wide scale customer acceptance. In order to reduce the size of the tank the team decided to redesign the system to chill the water to a lower temperature (38°F) and use a bleed back loop with a three-way thermostatic valve to deliver 58°F water to the panels. This reduced the required size of the tank to a maximum of 730 gallons for the case of a poorly insulated two story house in CZ15.

The team tested two different designs for the tank, a hard tank and a soft tank.

2.3.3.2 Hard tank

Figure 18: Fully Assembled Prototype Hard Tank

Photo Credit: Western Cooling Efficiency Center

The hard tank is intended for use outside the house and is designed to be sufficiently robust to withstand being sited in a high traffic area such as a side yard. The prototype has external dimensions of four feet by eight feet by four feet (height x length x width), with an internal volume of approximately 570 gallons. The walls are made from steel faced insulated panels with four inch thick isocyanurate foam insulation rated at R-31. A welded steel frame holds the panels together and provides stiffening against the hydrostatic forces. The base of the tank is also four inches of isocyanurate insulation. The tank is lined with 30 mil thick polyvinyl chloride (PVC). The prototype used a custom vinyl covered spa cover as a lid (a functionally similar cover could be made in bulk for a substantially lower cost) with an R value of 30.

The heat exchanger is a bare copper coil positioned at the top of the tank, which will allow the water to be cooled using natural convection, Figure 19.

After the first prototype was built, a design change was made so that the chilled tank water is not circulated through the panels in the cooling mode. Water from the panels is circulated in a closed loop through a PP (Polypropylene) coil heat exchanger immersed in the tank. In heating mode, the panel water circulates though a flat plate heat exchanger, the hot water is supplied from the gas water heater. This isolation of the panel water eliminates the possibility of contamination of the domestic hot water by the tank water in cooling mode and isolates the potable water from the circulating space heating water in the heating mode.

Figure 19: Storage Tank with Evaporator and Heat Exchanger Coil

Photo Credit: Gas Technology Institute

2.3.3.3 Soft tank

The soft tank design is intended for use inside the house, most likely in a basement. It consists of a cylindrical outer shell made of high tensile fabric, and an inner liner. On assembly, the space between the two is filled with 4 inches of insulating foam board. The soft tank therefore has the advantage of being able to be assembled on site, with the insulation being procured locally, thus reducing shipping costs.

The prototype was fabricated by American Solartechnics. The original plan was to build an eight foot tall and four foot wide inch diameter cylinder since a narrow cylinder meant there would be less occupied space; however this advantage came at the expense of a less stable tank. After consultation with American Solartechnics, the team decided to keep the height to width ratio low in order to yield a more stable design at the expense of a larger occupied area. Accordingly, the tank dimensions for this test were set at four feet in height and six feet in diameter. The heat exchanger, designed as a coil, was built at Beutler in Sacramento.

Figure 20: Prototype Soft Tank Undergoing Lab Testing

Photo Credit: Western Cooling Efficiency Center

2.4 Laboratory Testing

2.4.1 Tank

2.4.1.1 Thermal gain

The system is designed to deliver water at 38°F to the thermostatic mixing valve, and return water at 68°F to the tank. The thermal storage of the tank is thus 141,000 Btu (570 gal * 30°F * 8.3 Btu/gal/°F).

The prototype hard tank wall is four inch thick insulation with a stated R value of 31, resulting in k = 0.032 Btu/hr/ft^2. The surface area (including top and bottom) is 160 square feet. A simple 1-D analysis shows that the heat gain per hour in an ambient temperature of 120°F (with a water temperature of 38°F) will be approximately 425 Btu, with a resulting temperature rise of less than 0.1°F/hr. This calculation overestimates the heat loss by ignoring the shape factor of the tank (and treating the ground as being at ambient air temperature), but suggests that heat gain by the tank will not significantly affect the performance of the system.

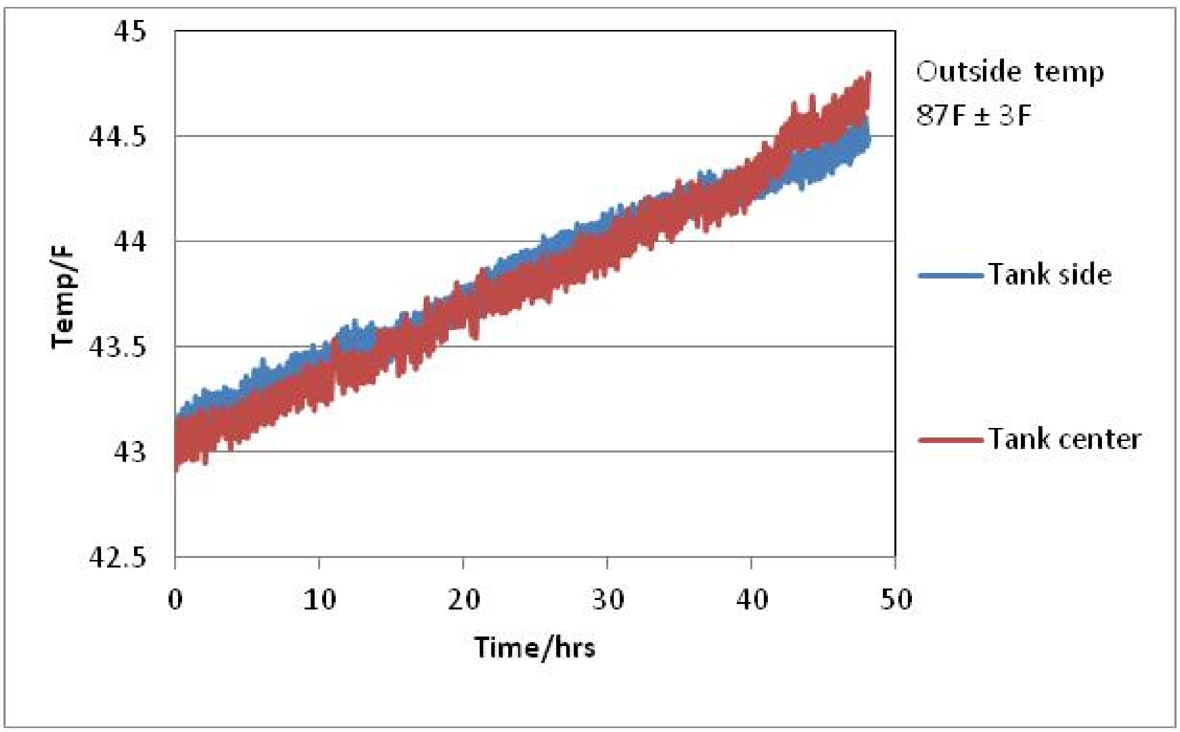
For the soft tank, the insulation is the same, but the surface area is 132 square feet, and the volume is 615 gallons, giving (again with an air temperature of 120°F and a tank temperature of 38°F) a heat gain of 350 Btu/hr, with a resulting temperature rise of less than 0.1°F.

The heat gain prediction is borne out in laboratory tests of the soft tank (Figure 21).

The tank was chilled to 43°F and monitored for 48 hours in an ambient temperature of 87°F. The temperature rise of approximately 0.03°F/hr was lower than the simple thermal modeling predicted.

As a comparison, during field tests the hard tank was monitored for 17 days in July (see Figure 53) and showed an average temperature increase of 0.07°F/hr.

Figure 21: Laboratory Test of Tank Thermal Gain

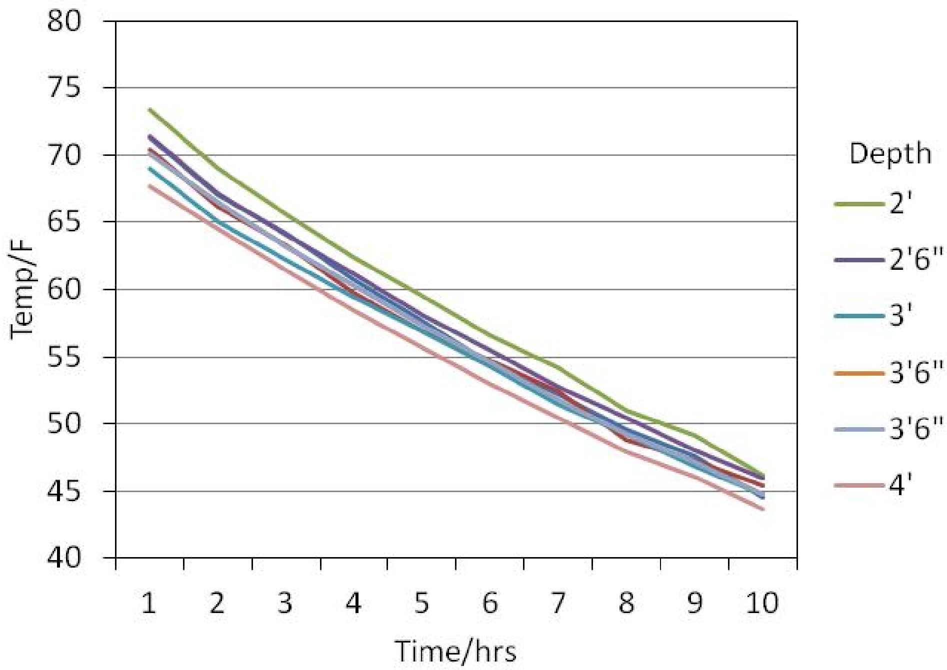


Source: Western Cooling Efficiency Center

2.4.1.2 Tank cooling

To keep the tank cost down and the system simple, the tank is designed to rely on natural convection to circulate the water during chilling. With the evaporator at the top of the tank, chilled water will tend to sink, allowing the water to be come into contact with the evaporator and the tank to cool without the need for mechanical circulation of the water. Figure 22 shows the results of tests on tank cooling. It is clear that the tank is cooled evenly down to the bottom, showing that mechanical circulation is not needed.

Figure 22: Tank Cooling Profile



Source: Western Cooling Efficiency Center

2.4.2 Panels

Laboratory testing of the panels was carried out by the Gas Technology Institute. The following section summarizes the results of the cooling tests. Table 13 and Table 14 summarize the results of cooling performance for the two panel designs tested. Table 15 shows results of measurements of stratification and thermal gradient in the test chamber. The details of the lab testing apparatus are covered in Chapter 3: Radiant Heating.

Table 13: Laboratory Test Results Serpentine Tube Panel

Test Number	Water Inlet Temp. (°F)	Water Flow Rate (gpm)	Chamber Temp (°F)	Avg Water Inlet Temp (°F)	Avg Water Outlet Temp (°F)	Avg Water Flow Rate (gpm)	Center of Panel Surface Temp (°F)	Average Chamber Temp (°F)	Average Room RH (%RH)	Heat Flux (Btu/hr/ft²)
1	58	0.5	70	58.09	58.97	0.513	63.38	70.03	38.02	-6.40
2	58	0.5	75	57.57	58.89	0.511	65.07	74.36	33.95	-9.62
3	58	0.5	80	57.43	59.50	0.498	68.18	80.27	32.24	-14.74
4	58	0.5	85	57.95	60.25	0.498	70.37	84.39	28.79	-16.43
5	50	0.5	78	50.13	52.81	0.495	63.61	77.51	27.74	-18.93
6	58	0.5	78	58.17	59.76	0.502	67.13	78.46	31.47	-11.40
7	68	0.5	78	68.00	68.72	0.497	72.47	78.17	33.89	-5.15
8	58	0.1	78	57.88	62.48	0.150	68.00	77.08	31.93	-9.79
9	58	0.2	78	58.32	61.38	0.205	67.74	77.46	37.56	-8.95
10	58	0.3	78	57.97	60.34	0.318	67.20	77.35	36.49	-10.76
11	58	0.4	78	58.24	60.16	0.406	66.93	77.37	35.98	-11.11
12	58	0.5	78	57.87	59.56	0.498	66.87	77.83	34.16	-12.01

Source: Gas Technology Institute

Table 14: Laboratory Test Results Manifold Panel

Test Number	Water Inlet Temp. (°F)	Water Flow Rate (gpm)	Chamber Temp (°F)	Avg Water Inlet Temp (°F)	Avg Water Outlet Temp (°F)	Avg Water Flow Rate (gpm)	Center of Panel Surface Temp (°F)	Average Chamber Temp (°F)	Average Room RH (%RH)	Heat Flux (Btu/hr/ft²)
1	58	0.5	70	57.71	58.18	0.492	71.84	71.27	33.83	-3.34
2	58	0.5	75							
3	58	0.5	80	58.74	59.62	0.504	80.13	79.65	31.74	-6.30
4	58	0.5	85	57.80	58.81	0.493	85.49	83.23	26.40	-7.14
5	50	0.5	78	49.96	52.39	0.518	62.26	77.42	26.65	-17.96
6	58	0.5	78	58.26	59.00	0.512	78.41	77.91	33.19	-5.42
7	68	0.5	78	68.35	68.74	0.505	77.65	77.08	29.16	-2.81
8	58	0.1	78	58.48	60.11	0.126	77.93	77.74	30.44	-2.93
9	58	0.2	78							
10	58	0.3	78	58.39	59.37	0.312	77.69	77.71	33.18	-4.34
11	58	0.4	78							
12	58	0.5	78	58.50	59.29	0.489	78.21	77.49	30.17	-5.53

Source: Gas Technology Institute

Table 15: Temperature Stratification Testing

								Vertical Location, ft								
Test Number	Avg Water Inlet Temp (°F)	Avg Water Outlet Temp (°F)	Avg Water Flow Rate (gpm)	Center of Panel Surface Temp (°F)	Average Chamber Temp (°F)	Average Room RH (%RH)	Heat Flux (Btu/hr/ft²)	0	1	2	3	4	5	6	7	8
6 Manifold Cooling	58.38	59.78	0.500	70.66	77.29	31.55	-9.93	76.80	77.52	77.56	77.40	77.48	77.56	77.48	77.49	77.19
6 Tube Cooling	58.16	59.78	0.506	67.05	77.25	31.35	-11.69	76.85	77.54	77.66	77.55	77.58	77.64	77.55	77.64	76.17
3h Tube Heating	115.72	112.21	0.323	104.36	68.81	36.75	16.20	68.23	67.54	67.74	67.83	68.11	68.49	68.76	69.30	74.32
3h Manifold Heating	115.47	109.77	0.305	106.78	68.17	33.28	24.76	66.02	66.33	66.73	67.26	67.82	68.42	68.71	69.52	74.69

Source: Gas Technology Institute

The following notes apply to raw data collection and data analysis:

1. The temperature and flow conditions for each test were the nominal test conditions. In some cases the actual room temperature, supply water temperature, or flow rates varied from these nominal values. The actual measurements were used in the analysis.
2. The average room temperature included all the wall surface and ambient air temperatures measured to better characterize the radiant environment.
3. Many of the planned cooling mode tests were conducted on the manifold panel and some of the planned heating mode tests were conducted before the leaking prototype panel could no longer be repaired and testing was abandoned.
4. MRT (mean radiant temperature) data were collected. There was no significant difference between the MRT temperature and the bare RTD (resistive temperature device) temperature near the MRT globe.
5. Tests were not randomized – they were performed in the order shown.

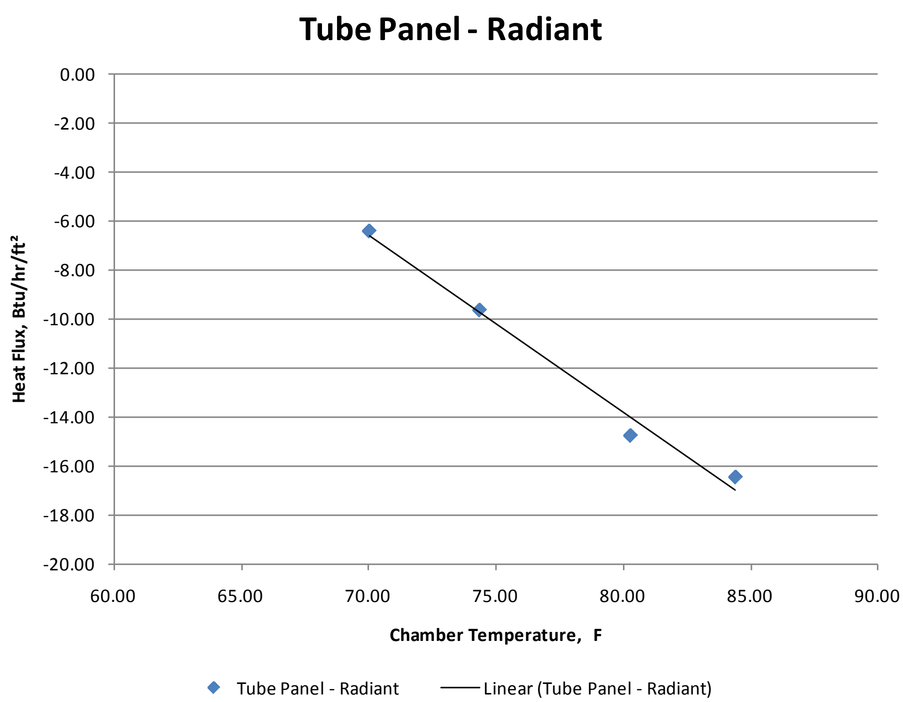
2.4.2.1 Panel test results

Figure 23, Figure 24, and Figure 25 summarize results of the cooling capacity of the serpentine (tube) panel as a function of various parameters.

In the cooling mode, flow rate is the most significant factor, followed by room temperature and then water inlet temperature. At 0.5 gallons per minute and 58°F delivered water temperature design conditions, the heat flux is -10 to -12 Btu/hr/ft^2 (absorbed heat is negative) at a 78°F room temperature. That equals -340 to -408 Btu/hr for a 34 square foot panel.

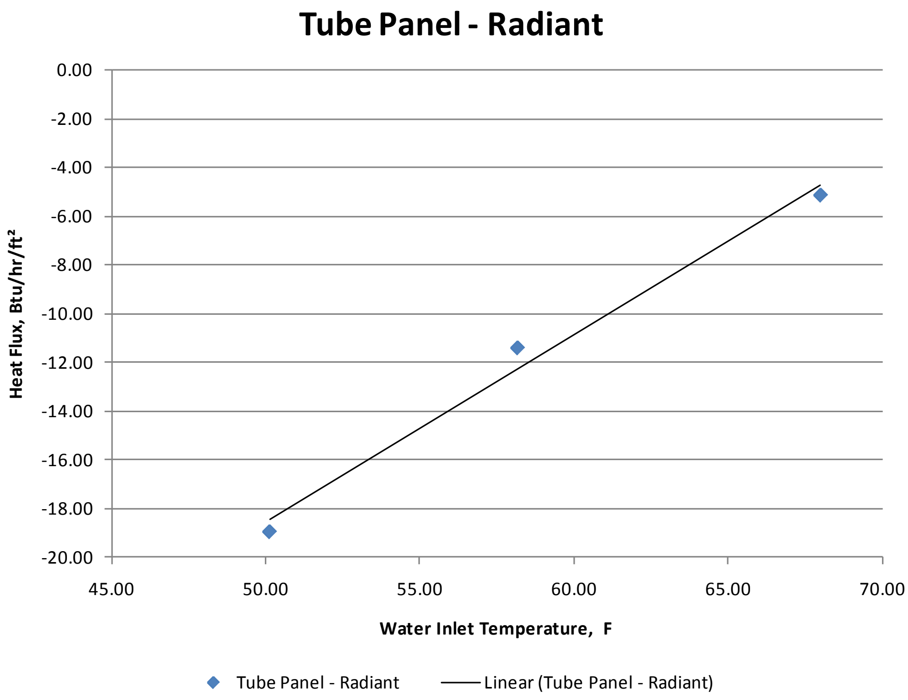
All of these tests were done with unpainted panels. Painting the panels with a flat white paint improved the cooling capacity by an average of 38 percent. For the field tests, the panels were painted white, and mounted in wood frames for aesthetic reasons.

Figure 23: Radiant Cooling Performance for the Serpentine Tube Panel with Varying Room Temperatures



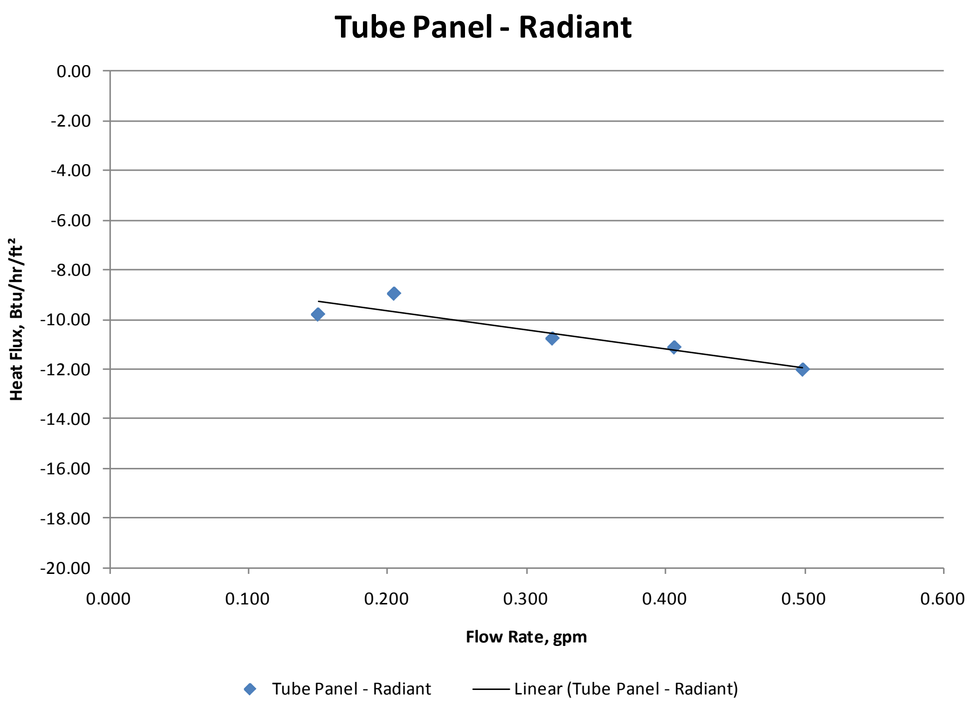
Source: Gas Technology Institute

Figure 24: Radiant Cooling Performance for the Serpentine Tube Panel with Varying Water Inlet Temperature



Source: Gas Technology Institute

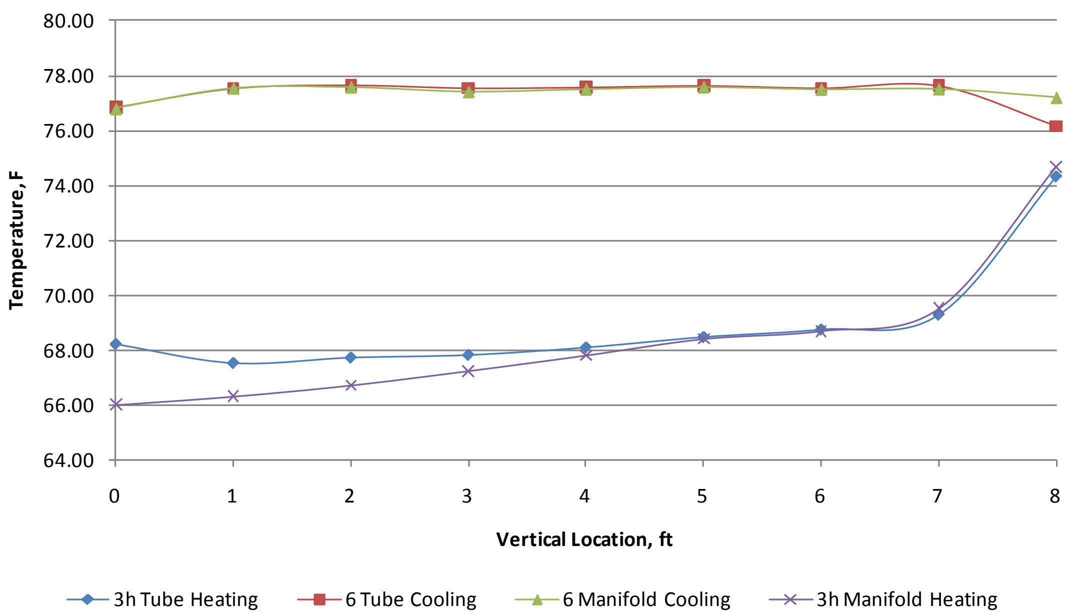
Figure 25: Radiant Cooling Heat Flux for Tube Panel with Varying Water Flow Rate



Source: Gas Technology Institute

Finally, the stratification tests show very even room temperature in a purely radiant environment with slight variation at the floor and at the ceiling. The radiant heating case shows the most increase above seven feet from the floor, rising four to six degrees F. With some convective air flow, it is anticipated that this variation will be reduced. Figure 26, below, shows this variation.

Figure 26: Temperature Stratification with Radiant Panels



Source: Gas Technology Institute

The Uponor panel sample was also tested in the lab for heating and cooling performance. The results are provided in Table 16, below. These results are consistent with performance information from the manufacturer: 16 Btu/hr/ft² for cooling at 58°F supply water temperature and 0.3 gallons per minute and 33 Btu/hr/ft² for heating at 120°F supply water temperature at 0.3 gallons per minute.

Table 16: Heating and Cooling Tests Uponor Panel

Water Inlet Temp. (°F)	Water Flow Rate (gpm)	Chamber Temp (°F)	Heat Flux (Btu/hr/ft²)
120	0.3	68	31.64
58	0.3	78	-16.01

Source: Gas Technology Institute

2.5 System Cost Analysis

The purpose of this project was to demonstrate the feasibility of an affordable radiant system with peak load shifting and controls designed to be a simple as possible. Affordability is addressed in this section.

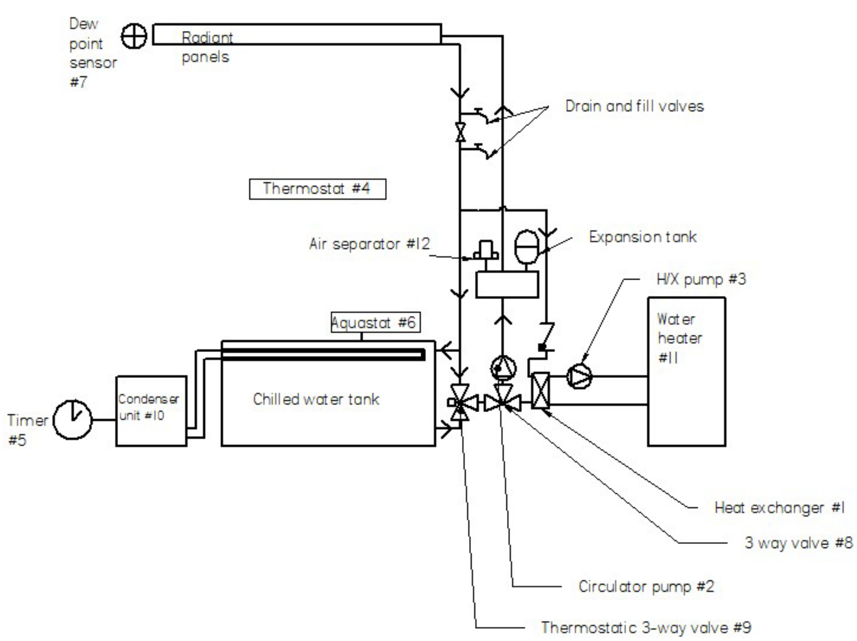
2.5.1 System Components

The system can broadly be divided into three sections: the controls, the storage tank and the radiant panels. As shown in Figure 27 the system consists of a circuit in which water is circulated through the ceiling panels. A three way valve (#8) switches the flow between two heat exchangers to set the system to either heating or cooling mode. In cooling mode the water flows through a water/water heat exchanger in the chilled water storage tank, and a thermostatic three way valve (#9) regulates the return temperature of the water to the panels. In heating mode, the water passes through a flat plate heat exchanger connected to the water heater (#1).

A dew point sensor (#7) switches the system off if condensation occurs on the panels.

The condenser is controlled by an aquastat to maintain the storage tank at the desired temperature during the cooling season, and a timer is preprogrammed to shut out the compressor during peak hours.

Figure 27: System Layout Schematic



Source: Gas Technology Institute

The components of the system are detailed in Table 17

Table 17: System Components

1	Heat exchanger	FlatPlate	GBE400H141
2	Circulator Pump	Taco	VDT0012
3	H/X Pump	Taco	006
4	Thermostat	Lux	TX9000TS
5	Timer switch	Intermatic	EH40
6	Aquastat	Honeywell	L4006A1959
7	Dew point sensor	Omega	MFR0122
8	3-way valve	Assured Automation	31D EV
9	Thermostatic 3 way valve	Leonard valve	LV-981-RF
10	Condenser unit	Carrier	CA16NA
11	Heater	AO Smith	GPHE50
12	Air separator	Taco	49-075T-1
13	24V transformer	White-Rodgers	90-T40F3

Source: Western Cooling Efficiency Center

2.5.2 Component Cost Analysis

2.5.2.1 Pumping components

Table 18: Cost of Pumping Components

Pumping system		
	Current	Future
Main Pump	$ 380	$ 250
Three way valve	$ 340	$ 230
Thermostatic three way valve	$ 500	$ 350
Water/water heat exchanger	$ 168	$ 120
Hot Water pump	$ 101	$ 80
Dew point sensor	$ 371	$ 250
Air eliminator	$ 77	$ 50
Thermostat	$ 60	$ 25
Additional tubing, fittings and i	$ 500	$ 400

Source: Western Cooling Efficiency Center

The current costs are those paid during the field tests, and the future costs are anticipated costs for bulk manufacture assuming a full market penetration of 20 percent, chosen by the authors.

2.5.2.2 Tank

The costs for the principal components of the hard tank are listed below (Table 19):

Table 19: Hard Tank Component Costs

Hard Tank		
	Current	Future
Steel frames	$ 369	$ 150
Panels	$ 480	$ 192
Cover	$ 650	$ 80
Liner	$ 354	$ 80
Bottom insulation	$ 100	$ 60
Coroplast	$ 100	$ 40
Coil	$ 1,100	$ 500
Heat exchange tubing	$ 150	$ 80
Incidentals	$ 200	$ 50
Materials Total	$ 3,503	$ 1,232
Labor	$ 4,500	$ 500
Overall total per tank	$ 8,003	$ 1,732

Source: Western Cooling Efficiency Center

For the soft tank, the costs are as below (Table 20):

Table 20: Soft Tank Component Costs

	Individual cost/$	Projected bulk cost/$
Tank fabric	745	600
Insulation	175	130
Plumbing/piping	200	100
Heat exchanger	500	200
Overall Total	1620	1030

Source: Western Cooling Efficiency Center

2.5.2.3 Panels

It is not terribly useful to use the cost of the network prototype as an indicator of the likely production cost of a panel of this type. The main cost of the prototype comes from the copper manifold, which will not be part of the final product.

As part of the cost analysis the team hired a plastics fabrication firm (PolyFab of Wilmington, MA) to investigate the feasibility of making the tubing mat, and the cost of doing so. The results of their analysis were that the cost of tooling required to make the network from extruded tubing would be prohibitive, for the quantities needed for the field testing part of this project. A method was designed for making the network from a manifold with welded bosses and either push fit or welded connections to the cooling tubes. The unit cost of either of these methods, for the limited quantities required for field testing (approximately 200 units), cannot be brought below $140. The cost of the tube required for the serpentine design is less than $13 (retail), so the extra cost of the network design is not justified.

The aluminum sheet is handled differently for the two designs as discussed above. For the network design, the cost of the material is low ($3 per panel) but the labor cost is higher than the serpentine design. The serpentine design uses a press formed aluminum sheet, which was priced at approximately $100 each. For sufficiently large quantities, aluminum sheet for the serpentine design could be roll formed, which would significantly reduce the cost of producing this type of panel as the labor requirement is negligible once the cost of the tooling has been amortized. As for the tubing network, the cost of tooling for roll forming is not an economically viable option for this project.

After the prototyping phase of the project, an opportunity arose to partner with Uponor and use a panel designed by them for the field test phase of the project. These panels are part of Uponor's European product line and consist of 15mm (0.59 in) thick gypsum panels with 10 mm (0.39 in) outside diameter PEX tubing inserted in routed channels. The panels are backed with 27 mm (1.06 in) thick expanded polystyrene (EPS) insulation. The design of these panels allows them to be installed similarly to ordinary drywall, which would make them particularly suitable for new construction projects, as the cost for installation and finishing would be partially offset by the savings from the drywall. It was decided to use the Uponor panels, along with the serpentine WCEC (Western Cooling Efficiency Center) designed panels, for the field tests. The cost model in Table 21 is for these two designs.

Table 21: Panel Cost Model

Panel costs			
Uponor		Current	Future
	Tubing	$ 720	$ 300
	Drywall	$ 230	$ 150
	EPS	$ 600	$ 300
	Assembly	$ 250	$ 150
	Installation	$ 2,000	$ 2,000
	Manifolds	$ 1,000	$ 300
WCEC			
	Panel	$ 5,000	$ 1,000
	Tubing	$ 580	$ 300
	Insulation	$ 290	$ 100
	Frame	$ 5,400	$ 1,000
	Assembly and painting	$ 5,000	$ 500
	Installation	$ 1,000	$ 500

Source: Western Cooling Efficiency Center

These costs are based on 1000 square foot of panels. The current costs are those paid during the field tests, and the future costs are anticipated costs for bulk manufacture.

Table 22 provided total estimated system cost for the hard tank and the two panel options:

Table 22: System Cost

Component Cost	Uponor, $		WCEC Prototype, $	
	Current	Future	Current	Future
Pumps and controls	2497	1755	2497	1755
Soft tank	8003	1732	8003	1732
Panels	4808	3200	17270	3400
Total	15308	6687	27770	6887

Source: Western Cooling Efficiency Center

2.6 Field Tests

2.6.1 Purpose

The purpose of the field tests was twofold:

- To demonstrate the viability of the combined heating/cooling radiant system in residential applications in order to further the project goal of bringing the system to market.
- To gather detailed information on the performance of all aspects of the system under *normal* use conditions. Performance of the panel design, chilled water storage system, off-peak vapor compression cooling system, gas-fired hot water heating system, and associated pumps, valves, and controls was investigated. The field test allowed the research team to assess the suitability and operation of the various components, as well as of the system as a whole, and to determine what modifications may be needed to move forward into a commercial phase of the project.

2.6.2 Locations

Two systems were installed. Due to the downturn in the residential building industry, both systems were installed as retrofits. This will affect the installation costs of the test systems compared to installation in new construction, but the information gathered allows a determination of the additional costs or savings that would be incurred in new construction.

Both locations were in SMUD territory in Sacramento and in California Building Zone 12 (see Figure 28).

The first site is on Grandstaff Drive (will be referred to as Grandstaff) and has the following characteristics:

- Single story
- Slab on grade construction
- Three bedrooms
- 1 ½ baths
- Two car attached garage
- Built in 1972
- Approximately 1000 square ft.

Figure 28: California Climate Zones

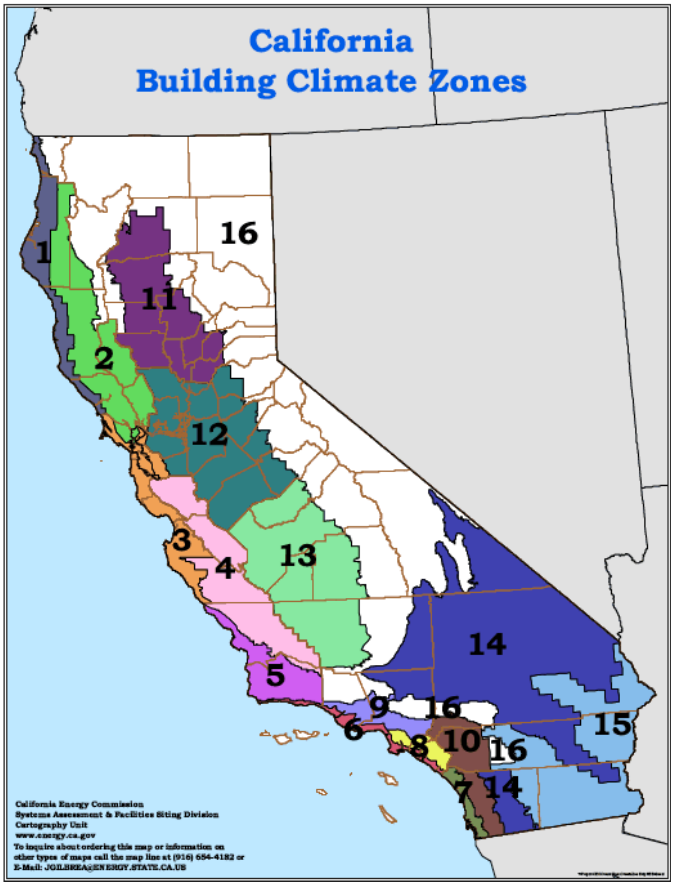


Photo Credit: California Energy Commission

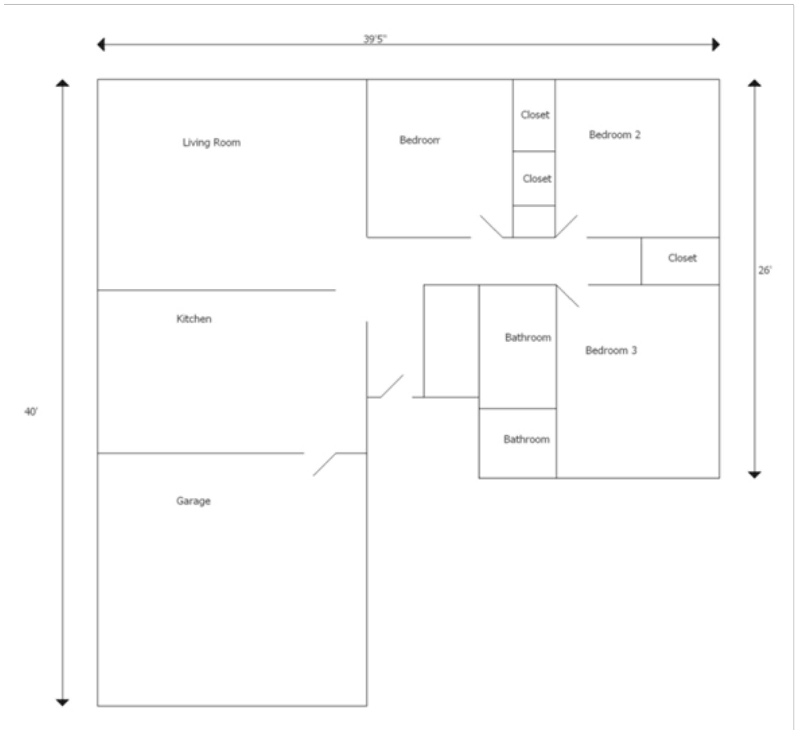
Figure 29 shows the Grandstaff site.

Figure 29: Grandstaff Field Test House

Photo Credit: Gas Technology Institute

Figure 30 shows the Grandstaff house floor plan.

Figure 30: Grandstaff House Floor plan



Source: Gas Technology Institute

The second field test site is on 6th Avenue (will be referred to as 6th Avenue) and has the following characteristics:

- Single story
- Crawl space construction
- Three bedrooms
- 1 ½ baths
- Detached garage
- Balloon framing (wall cavities open to the attic)
- Built in 1930
- Approximately 1000 square ft.

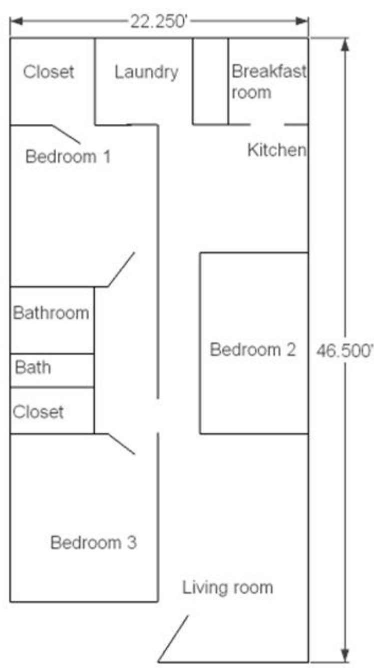
Figure 31 shows the 6th Avenue house.

Figure 31: 6th Avenue Field Test House

Photo Credit: Gas Technology Institute

Figure 32 shows the 6th Avenue floor plan.

Figure 32: 6th Avenue Floor Plan



Source: Western Cooling Efficiency Center

2.7 Installation

The critical path for installation was:

- Site visit to measure dimensions and plan layout
- Assemble system components
- Install system and controls
- Install sensors and data acquisition hardware
- Commission system

The systems were installed by Beutler Heating and Air. Beutler also conducted lab testing for the storage tank component of the system, providing continuity between the lab and field tests and ensuring that the install team was familiar with the system.

Two radiant panels were used in the field test. The first was a prototype panel (Serpentine Tube Panel); and the second was a commercial product sold by Uponor in Europe (Uponor Panel). The details of the two panel designs are as follows:

Figure 33 shows the serpentine (tube) panel. The characteristics are:

- Size – nominal 4 ft. x 8 ft. (actual 48.25 inches by 104.00 inches)
- Aluminum thickness – 32 mil (.032 inches), painted flat white
- Fiberglass board thickness and density – 1 inch, 7 pounds per cubic foot density
- Tubing material, size, layout – HDPE, serpentine layout with tubes 6 inches apart

Figure 33: Serpentine Tube Panel (Lab Setting)

Photo Credit: Gas Technology Institute

Figure 34 shows the Uponor Panel. The product characteristics are:

- Size – 500 mm x 1200 mm. (19.69 in x 47.24 in)
- Drywall bottom layer 15 mm (0.59 in) thick.
- 10 mm (0.39 in) PEX tubing in a channel cut into the drywall
- 27 mm (1.06 in) EPS foam insulation on the upper surface
- Mounting method – screw into ceiling joists or furring strips – metric size requires some framing
- Design issues – commercial product in Europe

Figure 34: Uponor Panel Close Up

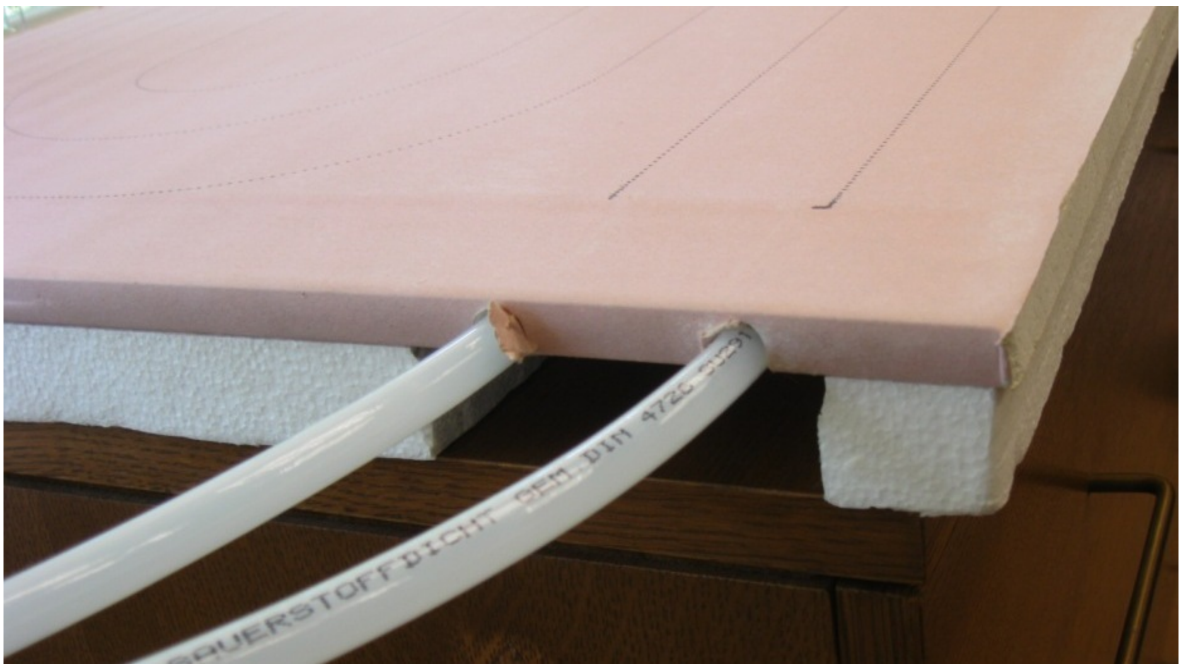
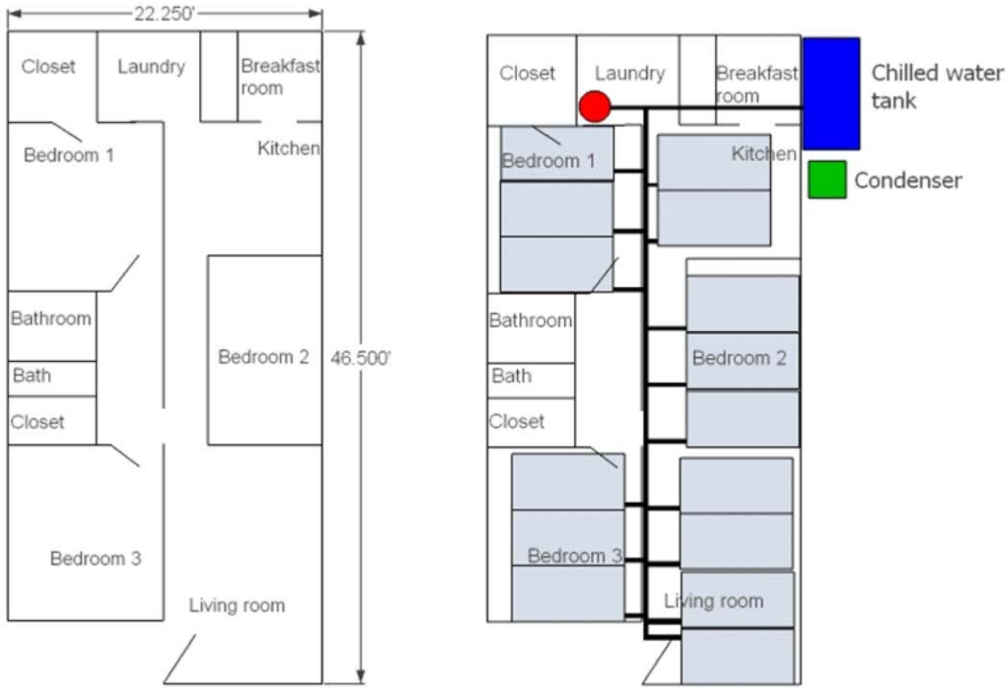


Photo Credit: Gas Technology Institute

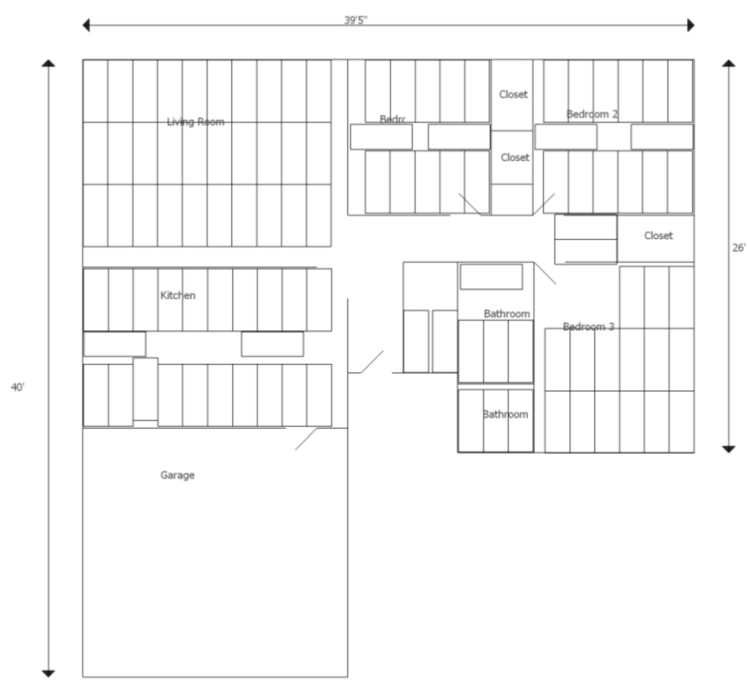
The Grandstaff house was fitted with Uponor panels in 18 circuits and the 6th Avenue house was fitted with serpentine panels. Figure 35 shows the 6th Avenue house panel layout and Figure 36 shows the Grandstaff house panel layout.

Figure 35: 6th Avenue House Panel Layout



Source: Western Cooling Efficiency Center

Figure 36: Grandstaff House Panel Layout

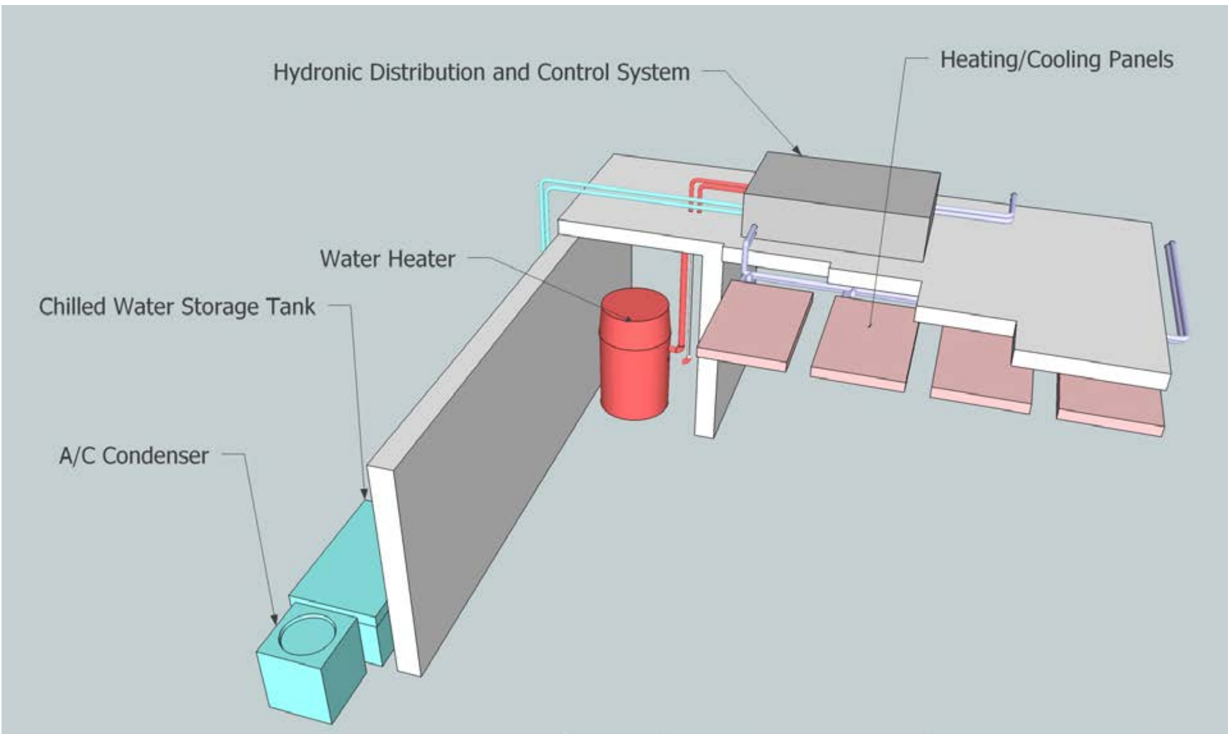


Source: Beutler Corporation

The hydronic system was designed to provide the water flow rate for adequate cooling at the design point of 15 Btu/hr/ft^2. The heating capacity at the same flow rate was calculated at 1.5 to 2 times the cooling capacity because of the larger driving temperature difference, so it was not necessary to adjust the water flow rate between heating and cooling modes. The 6th Avenue heating water flow rate was set at 4.8 gallons per minute for the whole house with 15 panels connected in parallel to manifolds. The flow rate for the Grandstaff house was set at 6.5 gallons per minute for the whole house with 88 panels running in 18 circuits. The chilled water tank supplied water at 58°F and the Vertex water heater was set at 120°F to provide the capacity required for the test.

Figure 37 shows the layout of the radiant heating and cooling system in a 3-D sketch. Note the position of the chilled water storage tank next to the A/C condenser and the routing of the plumbing to the hydronic distribution and control system.

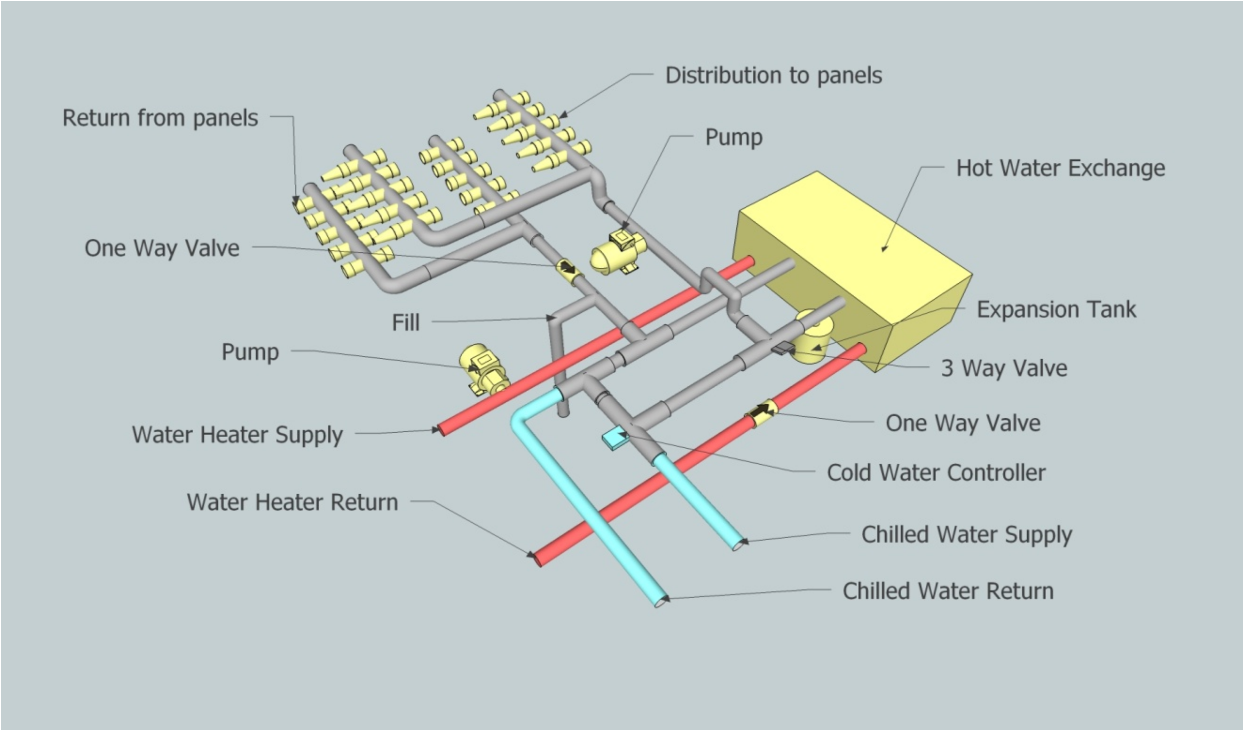
Figure 37: Schematic of Radiant Heating and Cooling System



Source: Gas Technology Institute

Figure 38 shows a schematic of the distribution and control system in the Grandstaff house in a 3-D sketch. Note the manifold for the 18 separate hydronic loops and the 3-way valve for switchover from heating to cooling.

Figure 38: Schematic of Hydronic Distribution and Control System



Source: Gas Technology Institute

Figure 39 through Figure 45, below, show various components of the system before and during installation.

Figure 39: Uponor Panels Installed In the Grandstaff House (Prior To Finishing)

Photo Credit: Gas Technology Institute

Figure 40: Uponor Piping Manifold in Grandstaff House

Photo Credit: Gas Technology Institute

Figure 41: Chilled Water Storage Tank in Grandstaff House

Photo Credit: Gas Technology Institute

Figure 42: Hydronic Control System in Grandstaff House

Photo Credit: Gas Technology Institute

Figure 43: High Efficiency Water Heater Installed In Grandstaff House

Photo Credit: Gas Technology Institute

Figure 44: Tubing Installation in the Attic of the 6th Avenue House

Photo Credit: Gas Technology Institute

Figure 45: Radiant Panels Installed In the 6th Avenue House

Photo Credit: Gas Technology Institute

Installation of the panels, hydronic system, and data acquisition system was followed by application of measured home performance thermal envelope improvements by Chitwood and Associates. See Figure 46 and Figure 47 for photos. That work is summarized in Chapter 5.

Figure 46: Measured Home Performance Installation in Grandstaff House

Photo Credit: Gas Technology Institute

Figure 47: Measured Home Performance Installation in 6th Avenue House

Photo Credit: Gas Technology Institute

2.7.1 Monitoring

System performance was monitored. Occupant comfort was self-reported by participants.

Data acquisition used a DataTaker DT85 to log the readings from all the sensors. The DT85 uses a built in cellular modem to allow remote monitoring and downloading of data. The remote monitoring reduces the need for frequent site visits.

Monitoring was broken down into three areas:

1. Energy: power consumption was monitored using Dent Powerscouts. The power consumption for the compressor and water pump were measured separately. Electrical power was measured using voltage and current pickups attached to the power cables. The Powerscout communicates with the DT85 using the ModBus protocol. Gas usage in heating mode was computed by monitoring a supplemental gas meter with magnetic pick-up.
2. Air conditions: Thermocouples were used to monitor the air temperature, and capacitive sensors were used to monitor the humidity. There was one temperature and one humidity sensor in each room of the house, and one temperature and humidity sensor on the outside of the house. External sensors were mounted so as to avoid direct sunlight. All thermocouples and humidity sensors connect directly to the DT85 for logging.
3. Water conditions: Temperature and flow were monitored. The temperature of the water was measured at the top, bottom and middle of the tank, as well as at both the inlet and outlet of the panels using thermocouples. The flow rate was monitored using a paddle wheel-type water sensor. As with the thermocouples, the paddle wheel flow sensor is logged using the DT85.

Details are provided in Table 23 below:

Table 23: Details of Instrumentation

Measurement	Device	Accuracy
Temperature (air)	Omega EWS-RH	±1.2°F
Relative humidity	Omega EWS-RH	±3% RH
Temperature (water)	Shielded T-Type Thermocouple	±1°F
Water flow rate	Omega FP-5600	±1% FS, 200 pulses/gal.
Gas flow rate	AC-250 Diaphragm Meter	1cuft
Electric power	Dent Powerscout 3	±0.5%

Source: Western Cooling Efficiency Center

Data is logged once per minute and stored locally using the DT85. Data is uploaded to the WCEC ftp server daily at 6am, and is retained on the Datataker for backup.

2.7.1.1 Data analysis

The data collected was continuously monitored and evaluated throughout the field test in order to identify any problems as they arose. Full analysis of all the data collected was carried out at the end of the test.

2.7.1.2 Schedule

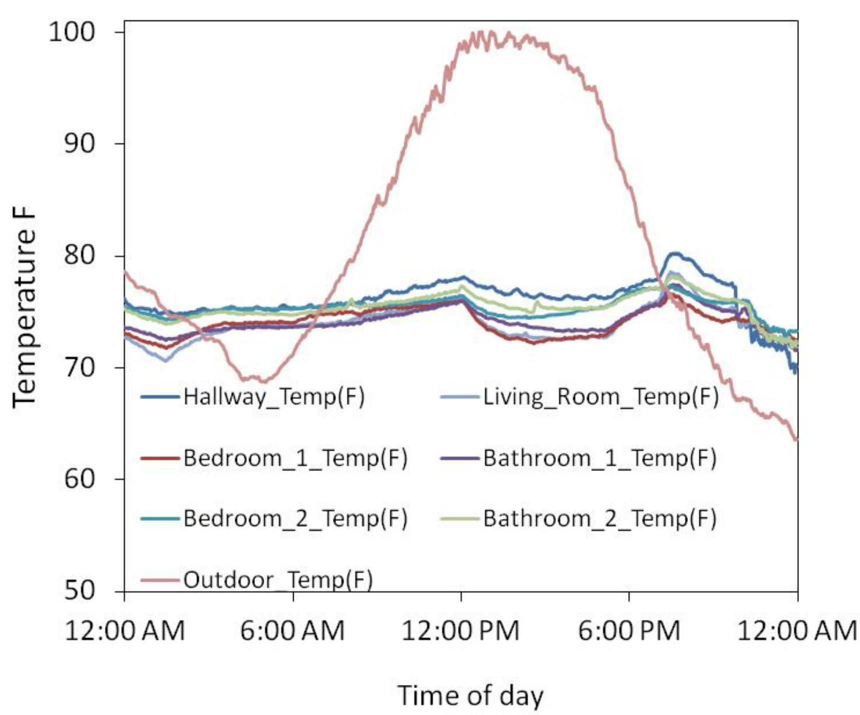
Installation was scheduled for the summer of 2011, with initial site visits carried out in March and April of 2012. The systems were fully commissioned in the summer to allow monitoring throughout the 2011/2012 heating season and the 2012 cooling season until the end of September.

2.8 Field Test Results

2.8.1 Daily Temperature Patterns

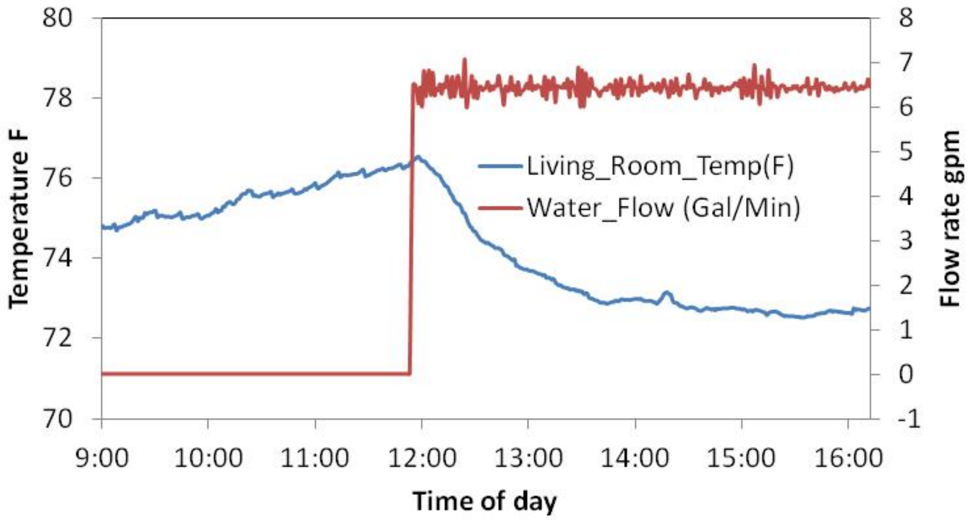
A typical temperature plot for Grandstaff Drive from June 17, 2012 is shown in Figure 48 below. The temperatures in the various rooms are seen to rise slowly through the morning until the temperature reaches the thermostat setpoint of 78°F and the cooling system switches on. Figure 49 shows the living room temperature and the chilled water flow rate during the middle of the day. Once the chilled water starts circulating through the panels the living room temperature can be seen to drop at an initial rate of approximately 3°F per hour.

Figure 48: Example Temperature Plot for Grandstaff Drive



Source: Western Cooling Efficiency Center

Figure 49: Living Room Temperature and Chilled Water Flow Rate

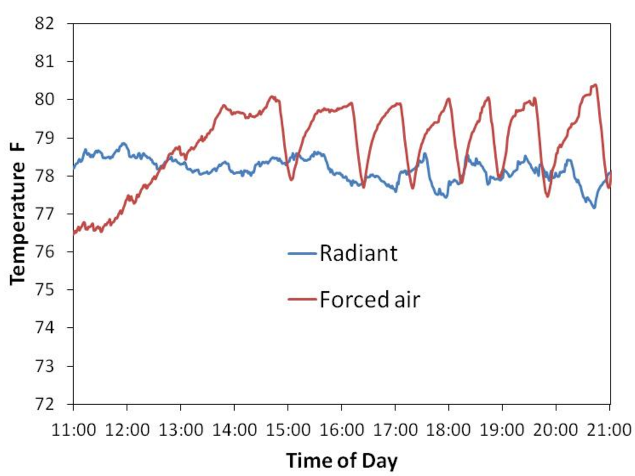


Source: Western Cooling Efficiency Center

2.8.2 Temperature Stability

Because the radiant system cools the environment by absorbing thermal energy radiated by surfaces in the house, rather than by blowing chilled air at a significantly lower temperature than the interior temperature as is the case for a forced air system, the temperature should be more stable. This expectation is borne out in Figure 50 which compares the temperature in the hallway when the house is cooled using the radiant system with the temperature when using the forced air system. The temperature measured when using the forced air system is seen to drop by ~ 2°F when the blower turns on. Occupants reported greater comfort with the radiant system than with the forced air system it replaced, citing the reduced temperature swings as one of the reasons.

Figure 50: Comparison of Radiant and Forced Air System Temperature Stability, Grandstaff Drive

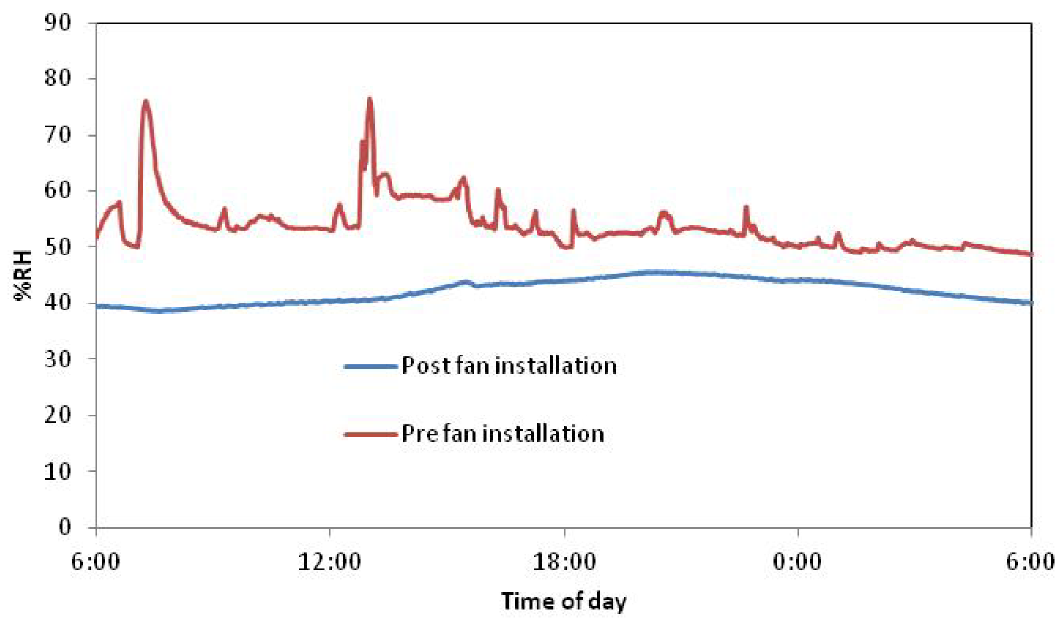


Source: Western Cooling Efficiency Center

2.8.3 Humidity Control

The Grandstaff Drive test site did not experience any condensation issues throughout the test period. This site had both kitchen and bathroom exhaust fans in place at the start of the test, whereas the 6th Avenue site did not. Initially severe problems were reported with condensation on the cooling panel in the bathroom. As part of the home performance improvements carried out during this project, a Panasonic WisperGreen fan was installed – this provides a constant ventilation rate of 20 cfm, rising to 80 cfm when the motion detector is triggered, i.e. when the bathroom is occupied. Figure 51 shows the bathroom humidity for two days before and after fan installation. Once the fan has been installed the spikes in humidity resulting from showers are almost undetectable, and no problems with condensation have been reported by the occupants.

Figure 51: Typical Bathroom Humidity Levels At 6th Ave Pre and Post Fan Installation

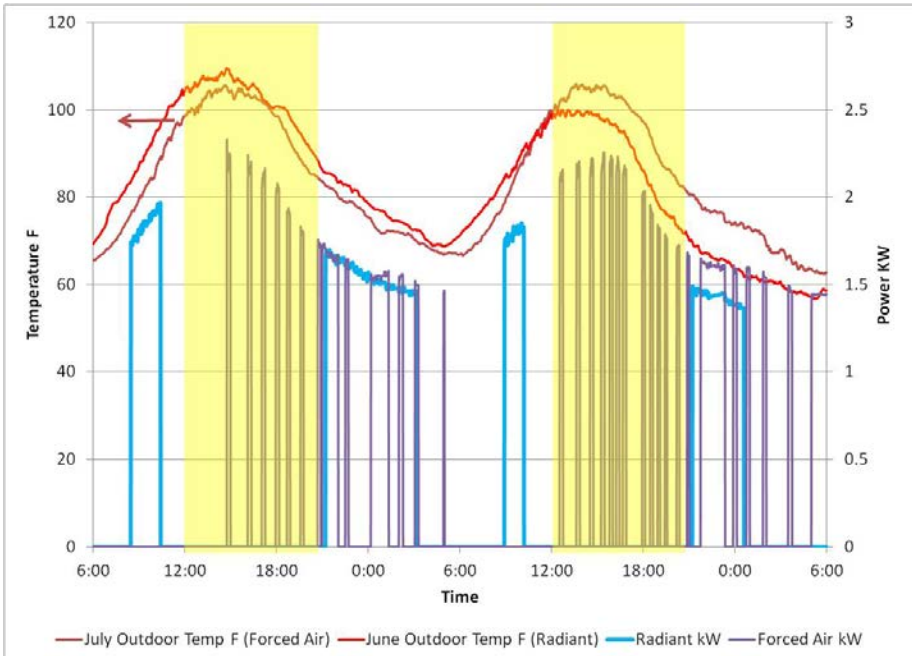


Source: Western Cooling Efficiency Center

2.8.4 Thermal Storage and Load Shifting

The thermal storage tank is designed to remove compressor load from peak hours. This aspect of the project has been totally successful. The size of the tank was calculated to ensure that the capacity would be sufficient to cover the load for the entire peak period, leaving the water pump as the only peak load. Figure 52 shows a comparison of the compressor power draw for the radiant and the forced air system at Grandstaff Drive. The radiant data is from June 17th and June 18th, 2012 and the forced air data is from July 21st and July 22nd. The dates were chosen as the outdoor peak temperature averaged 104 to 105 °F for the two days and the shape of the curve is very similar over the two time periods. The peak hours are shown by the yellow highlighted areas in Figure 52. Two features of the graph are significant. First, the bulk of the power used by the forced air system is during peak hours, whereas the radiant system draws no power during these periods. Second, the power draw of the forced air system increases with increasing temperature due to the higher lift required of the compressor, showing the additional savings of the radiant system due to running the compressor off peak when outdoor air temperatures are generally lower.

Figure 52: Comparison of Radiant and Forced Air Power Draw Over 2 Days

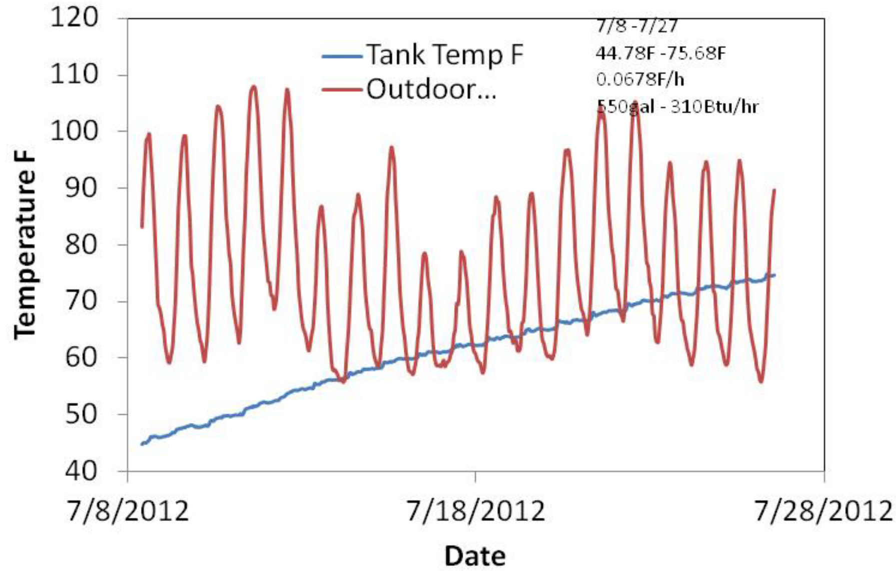


Source: Western Cooling Efficiency Center

2.8.5 Tank Heat Gain

Any thermal storage system will be subject to thermal gain, which will reduce the efficiency of the system as it is effectively an additional load. Figure 53 shows the thermal gain of the tank that was measured during July when the radiant system was off and the house was cooled using the forced air system. During this period the tank temperature rise averaged 0.068°F/hr, equivalent to a heat gain of 310 Btu/hr.

Figure 53: Tank Heat Gain with System Off

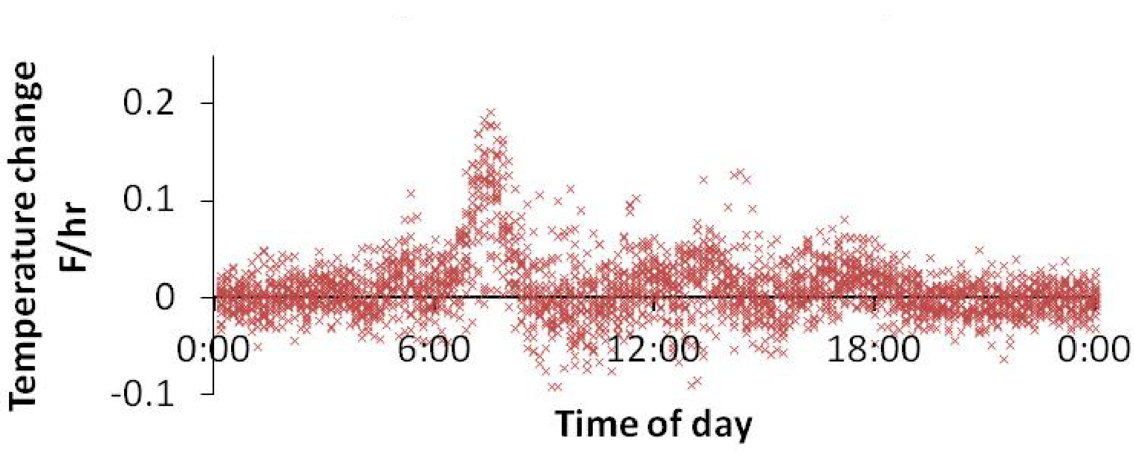


Source: Western Cooling Efficiency Center

2.8.6 Tank Location

Although the tank is primarily white in order to minimize heat gain, the location of the tank at the test site in Grandstaff Drive is such that it receives some direct sun in the early morning. The average hourly heat gain for July is plotted in Figure 54, where the effect of direct solar radiation is clearly seen. This graph is the result of converting the measured minute by minute temperature change to an hourly rate and superimposing data for each day

Figure 54: Tank Temperature Rise Plotted Vs. Time of Day with the System Off



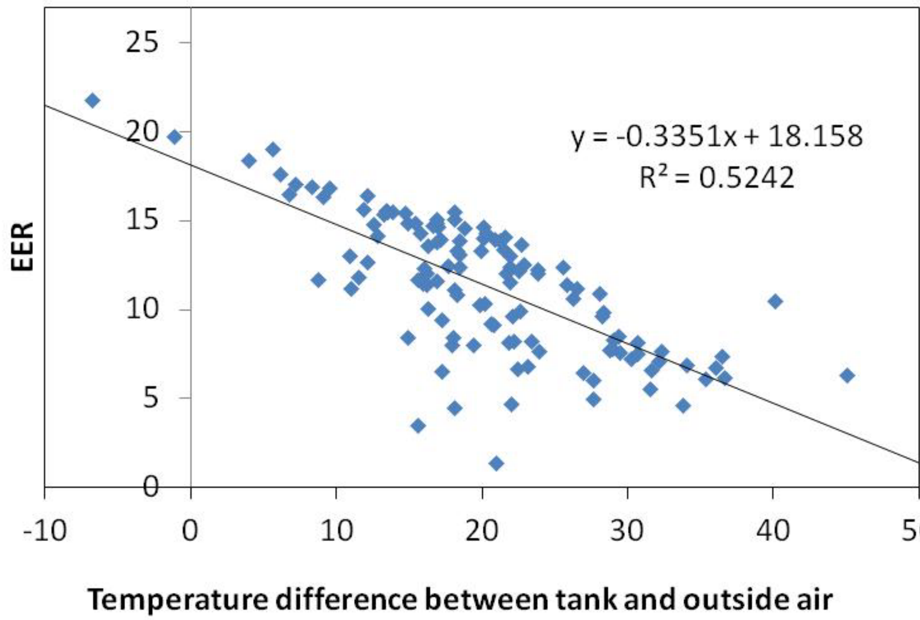
Source: Western Cooling Efficiency Center

2.8.7 Chiller Efficiency

The efficiency of the chiller is critical to the overall efficiency of the system. Although power savings are made when running the system at lower outdoor temperatures, it is clear from

Figure 55 that the efficiency of the chiller is significantly lower than the nominal SEER 16 rating of the condenser. The likely source of this low efficiency is the copper coil refrigerant to water heat exchanger, which has not been optimized. This is an area of the system that has room for improvement.

Figure 55: Chiller Efficiency as a Function of Tank Temperature and Outdoor Temperature

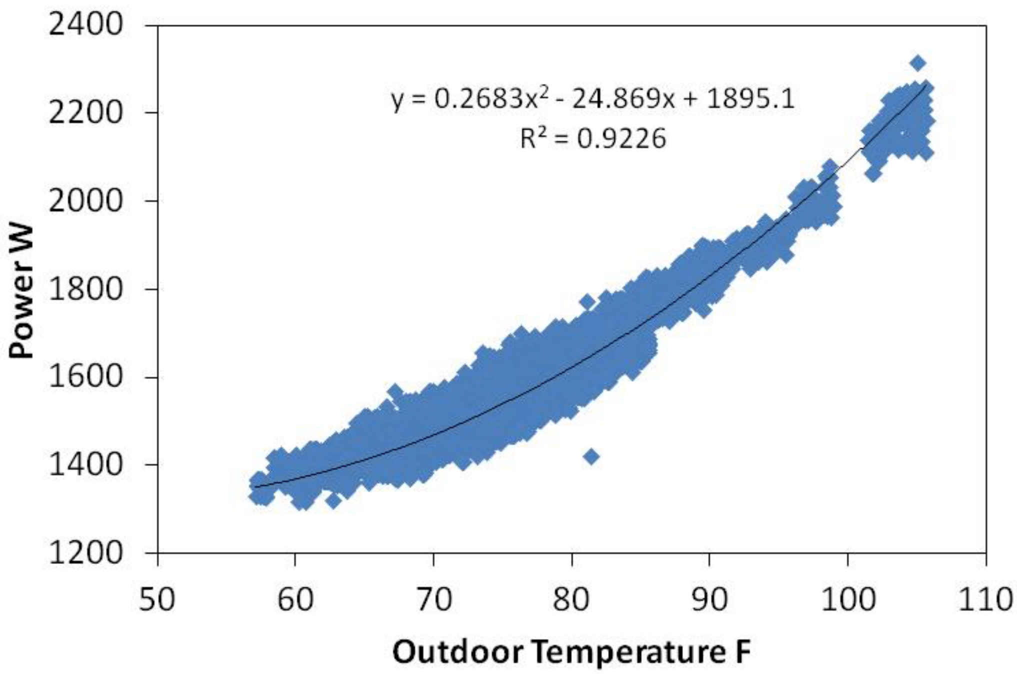


Source: Western Cooling Efficiency Center

2.8.8 Power Draw as a Function of Temperature

As seen in Figure 55, the power use of the compressor increases as the outdoor temperature increases. The full impact of this is shown in Figure 56, where the full dataset for Grandstaff Drive is plotted.

Figure 56: Compressor Power Draw as a Function of Outdoor Temperature



Source: Western Cooling Efficiency Center

The loss of efficiency of the condenser unit as a function of temperature allows a determination to be made of the energy savings due to the peak load shifting. By comparing the outdoor temperatures when the compressor is run to chill the tank, to the outdoor temperatures when cooling is delivered to the house (as determined by the times at which the water pump is running), the savings due to this effect can be calculated. A similar calculation has been carried out for 6th Avenue, and the results for both houses are shown in Table 24. The higher power savings at the Grandstaff Drive location are due to the fact that the 6th Ave site was unoccupied during the height of the summer, thereby missing the hottest days and the corresponding savings.

Table 24: Energy Use and Power Reduction Summary

	6th Avenue	Grandstaff
Peak power reduction	94%	94%
Energy savings	5%	19%

Source: Western Cooling Efficiency Center

2.8.9 Occupant Feedback

Occupants of both test houses expressed satisfaction with the system. Positive aspects were increased thermal comfort, greater temperature stability, and ease of use. Having cooled ceilings reduced the feeling of heat radiating from the ceiling late in the day and reduced temperature stratification. The only unsatisfying aspect reported was the inability of the system to adequately pull down the interior temperature if the house was allowed to become excessively warm. This is also a drawback of forced air systems, and underlines the need for proper temperature setback control.

2.9 Conclusions

The purpose of this project was to demonstrate the feasibility of a residential radiant heating/cooling system with peak load shifting. The systems used for the field tests successfully reduced peak loads by 95 percent and achieved total energy savings of 19 percent over the full cooling season, and can therefore be considered to have been completely successful. Anticipated problems were condensation on the panels and inadequate cooling, whether from lack of cooling power from the radiant surfaces or due to undersized chilled water storage. In both test houses the tank capacity was shown to be adequate, and the cooling capacity (Btu/hr/ft^2) of the panels was sufficient. Condensation was proved to be controllable with spot ventilation in the kitchens and bathrooms.

Further savings should be achievable by optimizing the tank heat exchanger, and by using an intelligent controller to optimize the times at which the compressor chills the water in the storage tank so that it is done at the coolest time of day and the tank is fully charged at the start of peak hours.

The panels used for the field test each have their own advantages. The serpentine panels can be installed without the need for finishing which reduces the cost for retrofits, whereas the Uponor panels produce a more traditional looking ceiling. Both designs performed well during lab and field tests.

The storage tank showed that it is possible to build a simple insulated tank with only minor mechanical agitation to circulate the water past the direct expansion coil. Using a separate water loop to circulate through the panels eliminates the possibility of contaminating the domestic water supply.

Although the system performed well, there is room for improvement. Cost estimates for the fully developed system (see page 41) are based on reduced component and assembly costs for the system as currently designed. It is likely that design changes could lead to reduced costs for the second generation system. Field test issues identified for both the heating and cooling system designs are covered in Section 3.3.4.

In order to fully develop the systems demonstrated in this project, a follow up study is recommended consisting of a broader test, targeting 20 houses. The target market would be houses undergoing deep retrofits – if the radiant panels are installed when a new ceiling is required; the cost of installing the panels is offset by saving the cost of the ceiling thereby reducing the marginal cost of the system. This would best be accomplished through close collaboration with one or more utility companies offering incentives for peak load reduction. Based on an incentive of $1200 per kW of peak load removed, this would equate to $2400 for the system as tested here, which would offset the cost of the storage tank. Refer to Appendix A for an analysis on ideal locations in Southern California for the radiant heating and cooling system.

CHAPTER 3: Radiant Heating

3.1 Purpose

The overall objective of the "Advanced Radiant HVAC Systems for California Homes" program is to integrate radiant cooling, heating, and related envelope systems and installation methods in California homes to produce significant energy savings as compared to traditional HVAC technologies and construction practices. This overall purpose of the heating tasks are to design and test heating system components to complement the cooling technology developed in the project so that a full HVAC system can be designed with minimal added cost. This chapter covers the design for the heating components of the system, integration of the components, laboratory and field testing, and utility bill cost savings.

3.1.1 Technology Evaluation

As mentioned in Chapter 2, the WCEC performed a residential building simulation using MICROPAS to further identify the heating and cooling loads for typical buildings in California Climate Zones 10 and 12 which are good locations for this system. Typical buildings, rather than Title 24 buildings, were used to represent the common retrofit case. Comparing cooling and heating loads in these estimates, the peak heating load was 1.5 to 2 times the peak cooling load for these houses. As such it was determined that the maximum heating capacity of the panels needed to be 1.5 to 2 times the maximum cooling capacity.

The bin analysis was extended to include several other factors. The load was reduced by 15.5 percent to account for duct losses that would not be realized with the hydronic system. This value was based on an analysis by WCEC. To account for the increased capacity needed to quickly recover from night setback, the loads were increased by a factor of 1.2. Further, a steady-state thermal efficiency of 90 percent was used to size the natural gas input capacity of the system. Table 25 provides the summary of that analysis. The maximum capacity for the heating equipment using the parameters identified was 37,100 Btu/hr.

Table 25: Capacity Analysis for Space Heating Equipment

	Single Story Zone 10	Two Story Zone 10	Single Story Zone 12	Two Story Zone 12
Max Hourly Load, kBtu	22.5	27.4	22.9	30.5
Adjusted for No Ducts (-15.5%)	19.0	23.1	19.4	27.8
Adjusted for Setback (x1.2)	22.8	27.7	23.3	33.4
Adjusted for Efficiency (÷90%) = Equipment Capacity	25.3	30.8	25.9	37.1

Source: Gas Technology Institute

3.1.2 Heat Source Options

There were several options available to provide the source of hydronic heating in the radiant system design: standard residential boilers, storage-type water heaters, tankless water heaters, or tankless water heaters with additional storage. Each had advantages and disadvantages in this application. The primary factors driving equipment selection were as follows:

1. Likelihood of no additional indoor floor area available for installing new equipment and water storage tank.
2. Installation issues associated with retrofitting atmospherically vented water heaters in some buildings.
3. Access to outside walls for PVC vented water heaters or direct vent water heaters.
4. Availability of ¾ inch gas lines for tankless water heaters over 80,000 Btu/hr input capacity.
5. Quick response to demand of less than 10,000 Btu/hr without short cycling.
6. Need to isolate the space heating system from the potable water heating system to avoid contamination with cooling loop water during changeover.

The best option for the retrofit case for the field test was to replace the existing water heater with a high-efficiency water heater with approximately an 80,000 Btu/hr capacity. Many water heaters offer separate side connections for the hydronic heating piping. A pump and heat exchanger was required to isolate the potable water from the chilled water loop. This option best addressed the factors above:

1. No additional floor area required.
2. No atmospheric venting required.
3. Access to an outside wall is required, but PVC vent pipes can be installed with lengths up to 100 feet (combined combustion air and vent) in many cases.
4. Only a ½ inch gas supply pipe is needed for the storage water heater up to 80,000 Btu/hr.
5. Stored water allows for small draws without firing the burner in many cases; short cycling is avoided producing less wear on the ignition system.
6. Isolation with a pump and heat exchanger is common practice for storage water heaters used for combo systems.

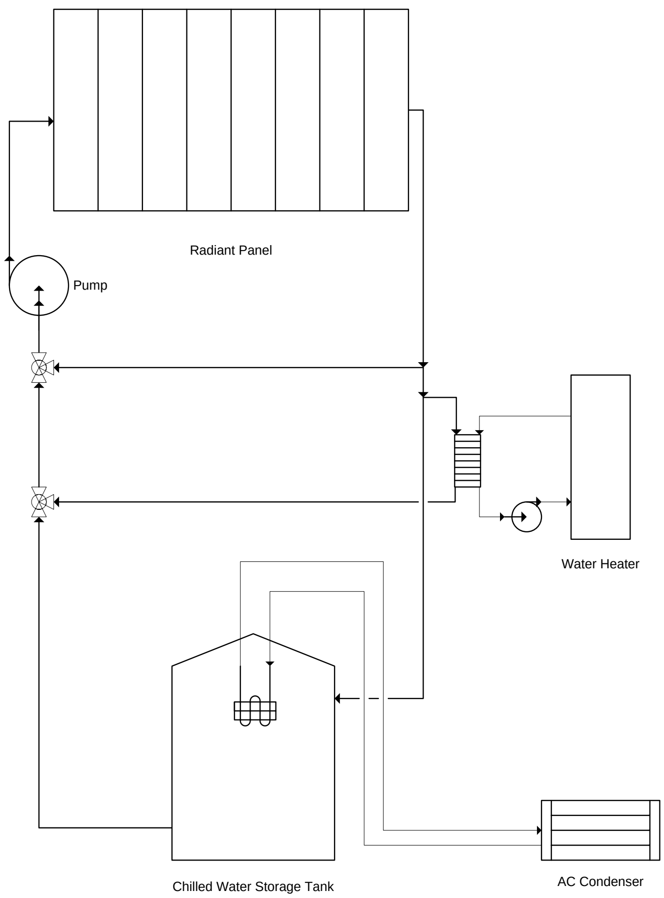
The A. O. Smith Vertex 76,000 Btu/hr 96 percent thermal efficiency water heater (Figure 43) was chosen for further evaluation.

The demand for residential hot water was not impacted by the selection of this unit for space heating. The system is sized to meet both loads simultaneously. In the laboratory test phase, the response of the water heater to a variety of simultaneous space heating and water heating loads was evaluated.

3.1.3 System Design Considerations for Integrating Heating and Cooling Functions

The radiant heating and cooling systems were designed to operate on a standard residential thermostat. When the homeowner switches from heating to cooling, three-way valves and pumps were designed to isolate the heating system from the cooling system and pump the desired fluid to the radiant panels. Figure 57 provides the current high-level schematic of that system. Note that some volume of water was transferred from one loop to the other during changeover, so the potable water system was required to be isolated from the hydronic heating and cooling loops to assure there was no contamination from the stored chilled water loop. Details of the design were evaluated in the laboratory testing phase.

Figure 57: Preliminary HVAC System Design



Source: Gas Technology Institute

3.2 Laboratory Testing of Radiant Heating Systems

3.2.1 Laboratory Test Apparatus

A new apparatus was built for the lab tests of the radiant panels in this project. The apparatus consisted of the following major components:

1. The chamber – a nominal 10 feet by 10 feet by 8 feet tall room built from lumber, insulation, and drywall. A 1 ton Sanyo mini-split heat pump was used for heating and cooling the space to precondition the room, and a series of tubes were placed on the floor and wall to provide the load for testing.
2. The HVAC system – this system was designed to both provide the load for testing and provide a controlled volume of heated and cooled water to the panels. Sub-components

included the water chiller, water heater, pumps and controls, hydronic distribution system, chiller heat exchanger and 300 gallon chilled water storage tank.

3. Instrumentation – A Lab View Fieldpoint data acquisition system was used. One set of modules was designed to operate pumps and the chiller in response to signals from the computer-based data acquisition system, and another set was used for collecting data on temperatures, flow volumes, and relative humidity in the chamber.

Several figures follow showing the lab test system components.

Figure 58: Test Chamber

Photo Credit: Gas Technology Institute

Figure 59: Chilled Water Storage Tank (Background), Plumbing System and Chilled Water Heat Exchanger (Foreground)

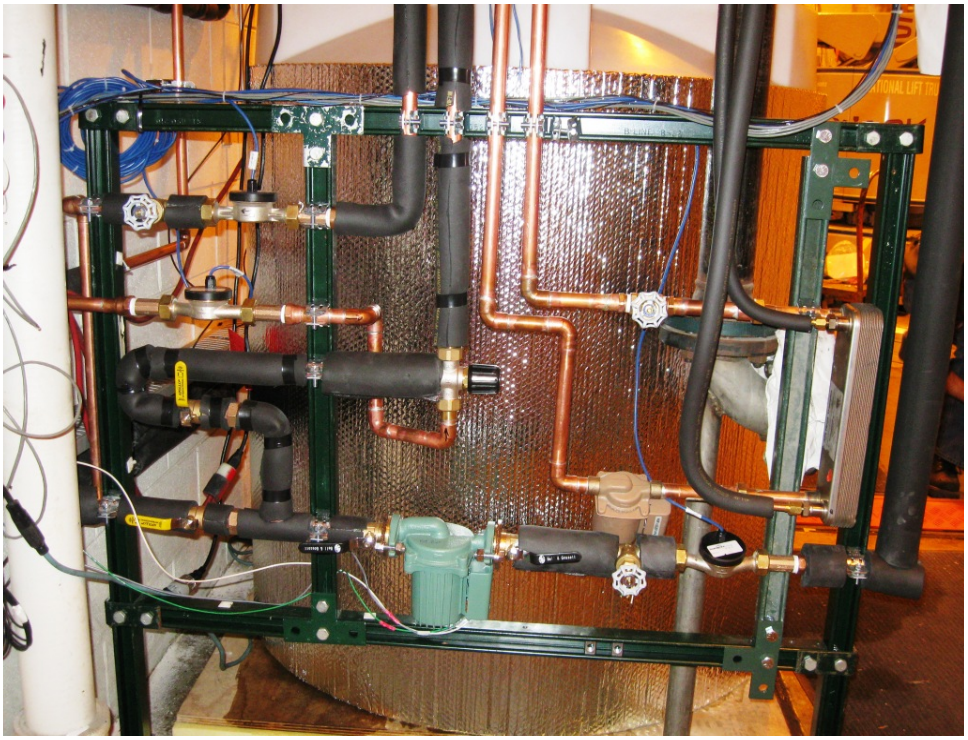


Photo Credit: Gas Technology Institute

Figure 60: Mini-Split Heat Pump (Foreground), Mini-Split Chiller for Chilled Water Production

Photo Credit: Gas Technology Institute

Figure 61: High Efficiency Water Heater with Side-Mounted Heat Exchanger for Panel Hot Water Supply

Photo Credit: Gas Technology Institute

Figure 62: MRT Globe, RH Sensor, Wall-Mounted Thermocouples, and Labview Fieldpoint Modules

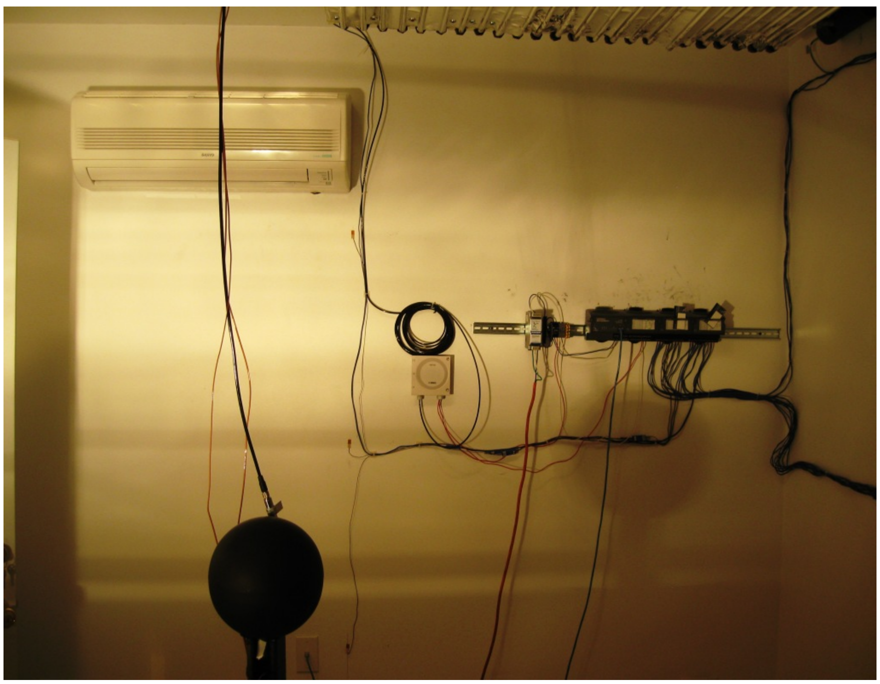


Photo Credit: Gas Technology Institute

Figure 63: 300 Pulses per Gallon Water Meter

Photo Credit: Gas Technology Institute

3.2.2 Radiant Panels Designs Tested in the Laboratory

Three radiant panel prototypes were tested. The serpentine tube panel design and Uponor panel design are covered in Section 2.7 and details are repeated here for completeness. The manifold panel design was also tested in the laboratory, but was not used in the field test.

The manifold panel uses a parallel tubing arrangement leading to headers on opposite sides. The tubing is sandwiched between a five thousandths of an inch aluminum sheet and a high density fiberglass board. The serpentine tube panel uses a single tube in a serpentine looping configuration sandwiched between a 32 thousandths of an inch aluminum sheet and high density fiberglass board. The third prototype was built using a commercial panel from the European company Uponor. The details of these designs are presented in the following section:

Manifold Panel (Figure 64):

- Size – nominal 4 ft. x 8 ft. (actual 48.75 inches by 97.125 inches)
- Aluminum thickness – 5 mil (.005 inches)
- Fiberglass board thickness and density – 1 inch, 7 lb. per cubic foot density
- Tubing material, size, layout – HDPE, manifold layout with tubes 2 inches apart
- Mounting method – screw into ceiling joists with large washers or screw into ceiling joists through manifold end framing
- Design issues – approximately 100 tube connections can leak

Figure 64: Manifold Panel

Photo Credit: Gas Technology Institute

Serpentine (Tube) Panel (Figure 65):

- Size – nominal 4 ft. by 8 ft. (actual 48.25 inches by 104.00 inches)
- Aluminum thickness – 32 mil (.032 inches)
- Fiberglass board thickness and density – 1 inch, 7 lb. per cubic ft. density
- Tubing material, size, layout – HDPE, serpentine layout with tubes 6 inches apart
- Mounting method – screw into ceiling joists; small washers may be needed
- Design issues – more robust design

Figure 65: Serpentine Tube Panel

Photo Credit: Gas Technology Institute

Uponor Panel

Figure 34 shows the Uponor panel that was tested. The specifications for this panel are presented below:

- Size – 500 mm x 1200 mm (19.69 inch by 47.24 inch).
- Drywall bottom layer 15 mm (0.59 in) thick.
- 10 mm (0.39 in) plastic tube (assumed to be PEX) in a channel cut into the drywall
- 27 mm (1.06 in) EPS foam insulation on the upper surface
- Mounting method – screw into ceiling joists or furring strips – metric size requires some framing
- Design issues – commercial product in Europe

3.2.3 Laboratory Test Plan

The following summary includes several changes that were made after the system shakedown was concluded.

The nominal operating conditions for the radiant heating system were 120°F delivered water temperature, 0.3 gallons per minute, and a chamber (room) temperature of 68°F. That condition was expected to produce a heat flux in the range of 15 to 20 Btu/hr/ft^2. Each of the main conditions was varied to determine the impact of a normal range of variation on the performance of the panel. The room temperature was varied from 60°F to 74°F in five increments. The supply water temperature was varied from 80°F to 140°F in five increments. The water flow rate was varied from 0.1 gallons per minute to 0.5 gallons per minute in five increments. These 15 tests, summarized below in Figure 66 were the basis for the radiant heating performance testing.

Figure 66: Heating Test Conditions

Test Number	Water Inlet Temp. (^oF)	Water Flow Rate (gpm)	Chamber Temp (^oF)
1h	120	0.3	60
2h	120	0.3	64
3h	120	0.3	68
4h	120	0.3	70
5h	120	0.3	74
6h	80	0.3	68
7h	100	0.3	68
3h	120	0.3	68
8h	130	0.3	68
9h	140	0.3	68
10h	120	0.1	68
11h	120	0.2	68
12h	120	0.3	68
13h	120	0.4	68
14h	120	0.5	68

Source: Gas Technology Institute

3.2.3.1 Convective Heating Tests

Convective heating tests were performed to determine if the radiant performance of the heating system could be improved by the addition of a fan in the conditioned space (such as a ceiling fan). For the serpentine tube panel, all 15 tests were re-run and for the manifold panel tests one, three, five, six, nine, 10, 12, and 14 were run with a 100 cfm blower operating in the room (mini-split indoor section shown in Figure 62). The Uponor panel was received too late to be tested for convective performance. Cooling tests were not run for the convective condition.

3.2.3.2 Stratification Tests

Four tests were run to measure the stratification of air temperatures from floor to ceiling as a way of assessing the comfort of the occupants. Test six in the cooling table was run for both panels, and test 3h in the heating table was run for both non-commercial panels. RTDs were mounted in a vertical line in the center of the room (not immediately below either panel) at one foot increments beginning at the floor and stopping at eight feet.

3.2.4 Raw Data Collected

Data was collected every five seconds for the tests identified and logged into the GTI network. The following data was collected for each test:

1. Time
2. Elapsed Time
3. Relative Humidity
4. Water Storage Chilled Water HX Inlet Temperature
5. Water Storage Chilled Water HX Outlet Temperature
6. Manifold Panel Surface Temp - South
7. Manifold Panel Surface Temp - Center
8. Manifold Panel Surface Temp - North
9. Tube Panel Surface Temp - South
10. Tube Panel Surface Temp - Center
11. Tube Panel Surface Temp - North
12. South Wall - 25"
13. South Wall - 49.75"
14. South Wall - 74.875"
15. West Wall - 25"
16. West Wall - 49.75"
17. West Wall - 74.875"
18. North Wall - 25"
19. North Wall - 49.75"
20. North Wall - 74.875'
21. East Wall - 25"
22. East Wall - 49.75"
23. East Wall - 74.875"
24. Ambient Room Temperature
25. MRT Globe Temperature
26. Manifold Panel Water Temperature Inlet
27. Manifold Panel Water Temperature Outlet
28. Tube Panel Water Temperature Inlet
29. Tube Panel Water Temperature Outlet
30. Water Heater HX Inlet
31. Water Heater HX Outlet

32. Water Heater Inlet
33. Water Heater Outlet
34. Chilled Water Storage Supply
35. Chilled Water Storage Return
36. Panel Return Water Flow Rate
37. Water Storage Supply Water Flow Rate
38. Panel Supply Water Flow Rate
39. Gas Meter (gas cubic ft.)
40. Panel Water Flow Rate (meter in chamber)

From these values, the following were calculated for each test:

1. Average Water Inlet Temp
2. Average Water Outlet Temp
3. Average Water Flow Rate
4. Center of Panel Surface Temp
5. Average Chamber Temp
6. Average Room RH
7. Panel Heat Flux

3.2.5 Data Analysis

Raw data was analyzed to identify the performance of the panels with the operating conditions identified. The results of each test are provided in the tables below.

Table 26: Radiant Heating Tube Panel

Test Number	Water Inlet Temp. (°F)	Water Flow Rate (gpm)	Chamber Temp (°F)	Avg Water Inlet Temp (°F)	Avg Water Outlet Temp (°F)	Avg Water Flow Rate (gpm)	Center of Panel Surface Temp (°F)	Average Chamber Temp (°F)	Average Room RH (%RH)	Heat Flux (Btu/hr/ft²)
1h	120	0.3	60	116.73	112.15	0.304	103.47	60.66	58.63	19.79
2h	120	0.3	64							
3h	120	0.3	68	116.39	112.35	0.303	104.60	65.99	47.51	17.56
4h	120	0.3	70							
5h	120	0.3	74	116.83	113.31	0.301	106.44	72.41	46.06	15.04
6h	80	0.3	68	74.41	74.12	0.335	73.69	66.75	42.75	1.41
7h	100	0.3	68							
3h	120	0.3	68	116.30	112.43	0.310	104.47	67.32	37.94	17.16
8h	130	0.3	68							
9h	140	0.3	68	133.83	128.62	0.302	118.42	68.53	49.57	22.47
10h	120	0.1	68	115.53	107.25	0.099	102.10	67.82	42.59	11.62
11h	120	0.2	68							
12h	120	0.3	68	116.30	112.43	0.310	104.47	67.32	37.94	17.16
13h	120	0.4	68							
14h	120	0.5	68	116.67	114.27	0.495	105.57	67.84	39.70	17.02

Source: Gas Technology Institute

Table 27: Manifold Panel (Test Stopped Due To Leakage)

Test Number	Water Inlet Temp. (^oF)	Water Flow Rate (gpm)	Chamber Temp (^oF)	Avg Water Inlet Temp (^oF)	Avg Water Outlet Temp (^oF)	Avg Water Flow Rate (gpm)	Center of Panel Surface Temp (^oF)	Average Chamber Temp (^oF)	Average Room RH (%RH)	Heat Flux (Btu/hr/ft²)
1h	120	0.3	60	115.32	109.67	0.315	106.41	60.21	67.13	25.33
2h	120	0.3	64							
3h	120	0.3	68	115.47	109.77	0.305	106.78	68.17	33.28	24.76
4h	120	0.3	70							
5h	120	0.3	74							
6h	80	0.3	68							
7h	100	0.3	68							
3h	120	0.3	68							
8h	130	0.3	68							
9h	140	0.3	68							
10h	120	0.1	68							
11h	120	0.2	68							
12h	120	0.3	68							
13h	120	0.4	68							
14h	120	0.5	68							

Source: Gas Technology Institute

Table 28: Convective Heating Tube Panel

Test Number	Water Inlet Temp. (^oF)	Water Flow Rate (gpm)	Chamber Temp (^oF)	Avg Water Inlet Temp (^oF)	Avg Water Outlet Temp (^oF)	Avg Water Flow Rate (gpm)	Center of Panel Surface Temp (^oF)	Average Chamber Temp (^oF)	Average Room RH (%RH)	Heat Flux (Btu/hr/ft²)
1h	120	0.3	60	115.87	107.97	0.277	96.49	60.88	30.51	28.21
2h	120	0.3	64	116.45	110.49	0.299	100.09	63.49	28.97	24.49
3h	120	0.3	68	117.01	111.45	0.314	101.40	65.98	28.09	24.91
4h	120	0.3	70	116.54	111.32	0.321	101.73	67.64	27.82	23.95
5h	120	0.3	74	116.41	111.27	0.296	102.95	73.44	41.90	21.59
6h	80	0.3	68	73.42	72.71	0.295	71.67	67.13	29.66	2.98
7h	100	0.3	68	107.03	101.90	0.295	93.06	66.56	27.82	21.65
3h	120	0.3	68	117.01	111.45	0.314	101.40	65.98	28.09	24.91
8h	130	0.3	68	125.05	117.85	0.293	106.74	66.33	27.46	30.16
9h	140	0.3	68	134.11	125.94	0.293	113.70	65.96	28.12	34.17
10h	120	0.1	68	114.60	102.81	0.111	95.87	66.24	37.52	18.83
11h	120	0.2	68	115.88	108.08	0.206	99.21	67.10	30.99	22.97
12h	120	0.3	68	116.11	110.84	0.300	101.00	66.95	38.83	22.77
13h	120	0.4	68	116.98	112.55	0.393	101.76	67.02	31.44	24.82
14h	120	0.5	68	115.42	112.13	0.508	101.14	67.43	41.87	23.82

Source: Gas Technology Institute

Table 29 Convective Heating Manifold Panel

Test Number	Water Inlet Temp. (^oF)	Water Flow Rate (gpm)	Chamber Temp (^oF)	Avg Water Inlet Temp (^oF)	Avg Water Outlet Temp (^oF)	Avg Water Flow Rate (gpm)	Center of Panel Surface Temp (^oF)	Average Chamber Temp (^oF)	Average Room RH (%RH)	Heat Flux (Btu/hr/ft²)
1h	120	0.3	60	114.55	107.06	0.307	103.91	62.36	60.44	32.73
2h	120	0.3	64							
3h	120	0.3	68	115.93	109.53	0.301	106.59	67.57	47.38	27.50
4h	120	0.3	70							
5h	120	0.3	74	116.15	110.38	0.303	107.90	73.04	39.89	24.90
6h	80	0.3	68	73.97	73.13	0.304	72.13	66.26	57.68	3.67
7h	100	0.3	68							
3h	120	0.3	68	115.93	109.53	0.301	106.59	67.57	47.38	27.50
8h	130	0.3	68							
9h	140	0.3	68	132.01	123.18	0.298	120.02	67.65	59.47	37.55
10h	120	0.1	68	114.82	100.83	0.099	100.99	67.33	46.19	19.85
11h	120	0.2	68							
12h	120	0.3	68	115.93	109.53	0.301	106.59	67.57	47.38	27.50
13h	120	0.4	68							
14h	120	0.5	68	116.61	112.31	0.511	107.68	67.77	35.81	31.34

Source: Gas Technology Institute

Table 30: Temperature Stratification Testing

								Vertical Location, ft								
Test Number	Avg Water Inlet Temp (^oF)	Avg Water Outlet Temp (^oF)	Avg Water Flow Rate (gpm)	Center of Panel Surface Temp (^oF)	Average Chamber Temp (^oF)	Average Room RH (%RH)	Heat Flux (Btu/hr/ft²)	0	1	2	3	4	5	6	7	8
6 Manifold Cooling	58.38	59.78	0.500	70.66	77.29	31.55	-9.93	76.80	77.52	77.56	77.40	77.48	77.56	77.48	77.49	77.19
6 Tube Cooling	58.16	59.78	0.506	67.05	77.25	31.35	-11.69	76.85	77.54	77.66	77.55	77.58	77.64	77.55	77.64	76.17
3h Tube Heating	115.72	112.21	0.323	104.36	68.81	36.75	16.20	68.23	67.54	67.74	67.83	68.11	68.49	68.76	69.30	74.32
3h Manifold Heating	115.47	109.77	0.305	106.78	68.17	33.28	24.76	66.02	66.33	66.73	67.26	67.82	68.42	68.71	69.52	74.69

Source: Gas Technology Institute

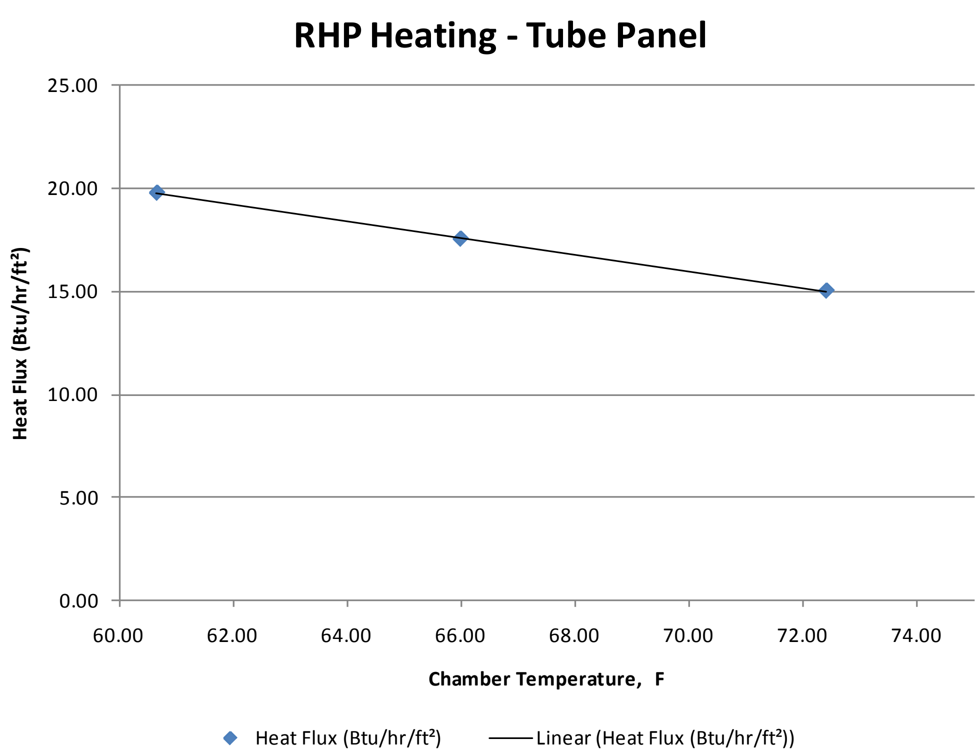
As mentioned previously, the following notes apply to raw data collection and data analysis:

1. The setpoints for each test were the nominal condition desired for each test, note that in some cases the actual room temperature, supply water temperature, or flow rates varied.
2. The average room temperature reported included all the wall and ambient air temperatures measured to better characterize the radiant environment.
3. MRT temperatures were collected. There was no significant difference between the MRT temperature and the bare RTD temperature near the MRT globe.
4. Tests were not randomized – they were performed in the order shown.

3.2.6 Panel Test Results

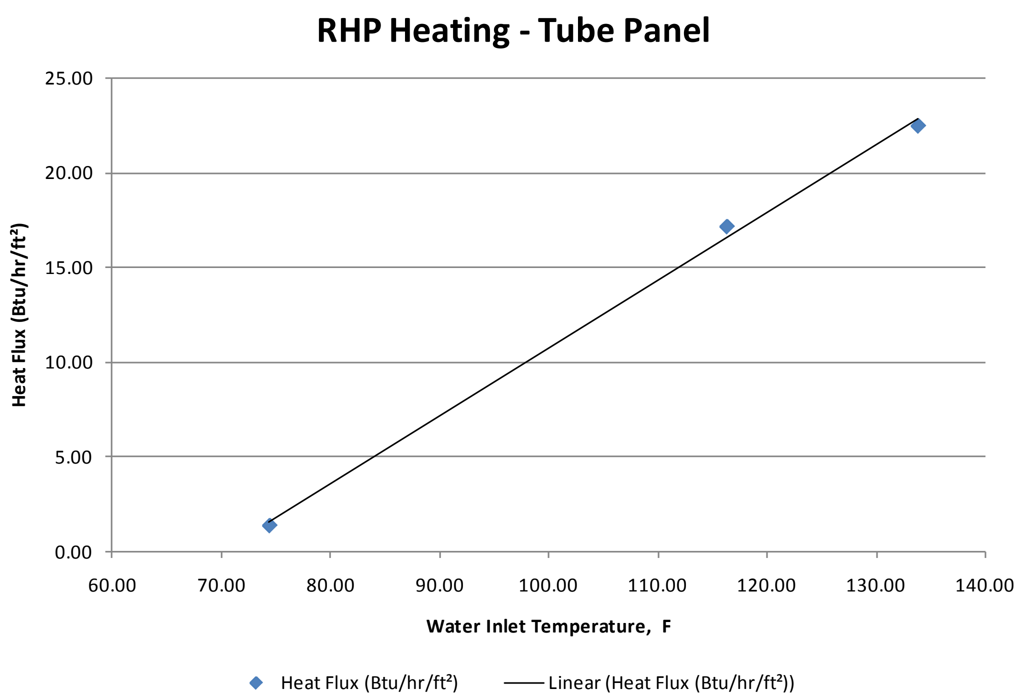
The manifold panel testing was stopped due to the large number of leaks in the design and the difficulty of repairing it at the ceiling height. The team decided that the manifold panel was not likely to stand up well in field testing, so the design was abandoned. The laboratory test results for the tube panel are plotted in the figures below. The Uponor production panel was received after the tests were complete; limited testing was performed on that panel at design conditions. Figure 67 shows the radiant heating performance with room temperature, Figure 68 shows the effect of inlet water temperature on heat flux, Figure 69 shows the effect of water flow rate on heat flux, Figure 70 shows the difference between convective and radiant heating performance, and Figure 71 shows the room temperature stratification.

Figure 67: Effect of Chamber Temperature on Heat Flux for Tube Panel in Radiant Heating Mode



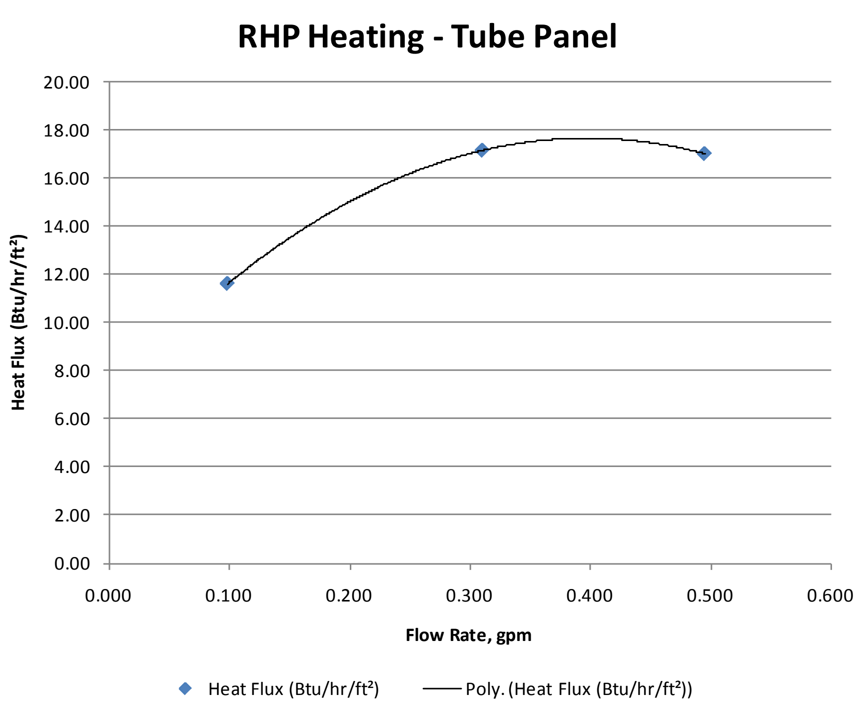
Source: Gas Technology Institute

Figure 68: Effect of Water Inlet Temperature on Heat Flux for the Tube Panel in Radiant Heating Mode



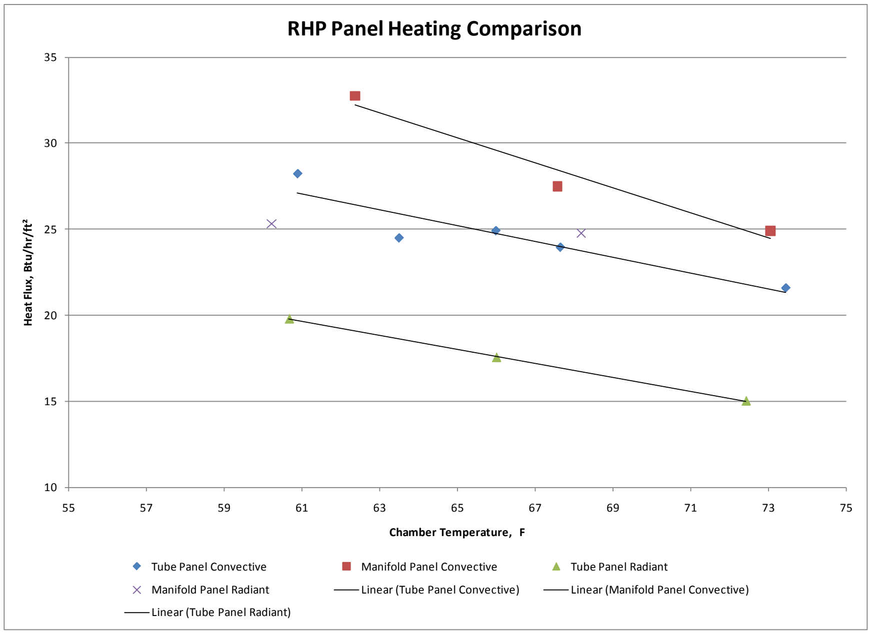
Source: Gas Technology Institute

Figure 69: Effect of Flow Rate on Heat Flux for the Tube Panel in Radiant Heating Mode



Source: Gas Technology Institute

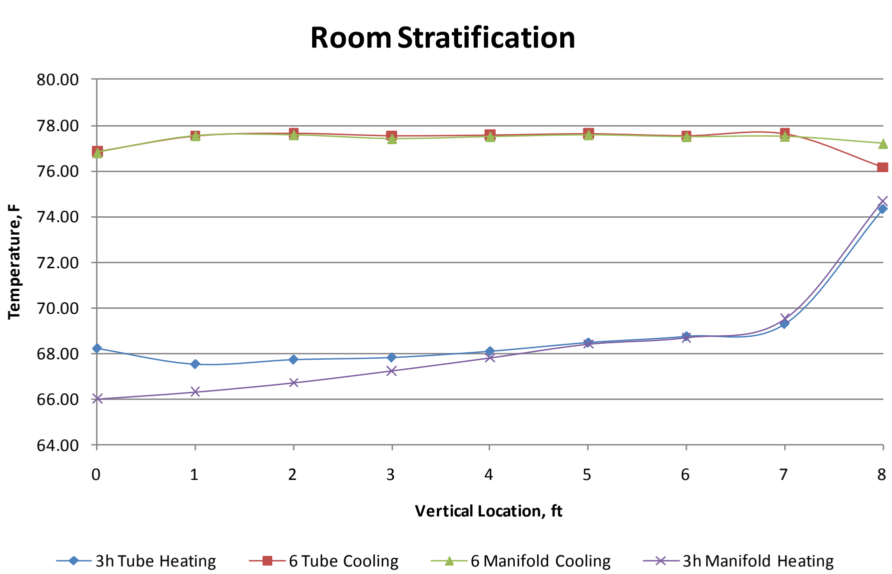
Figure 70: Convective vs. Radiant Heating Performance



Source: Gas Technology Institute

As mentioned in Chapter 2, the stratification tests showed very even room temperature in a purely radiant environment with slight variation at the floor and at the ceiling. The radiant heating case showed the most increase above seven feet from the floor, rising four to six degrees F. With some convective air flow, it was tested and verified that this variation would be reduced. Figure 71, below, shows this variation.

Figure 71: Temperature Stratification with Radiant Panels



Source: Gas Technology Institute

The radiant panels performed well in the laboratory testing. The heat flux on the heating and cooling season tests fell in line with the predicted values.

- For the radiant heating tests, the performance of the panel was most strongly dependent on the delivered water temperature, followed by the chamber temperature, and lastly, the flow rate. The flow rate impact was significant up to 0.3 gallons per minute and then leveled off. At 0.3 gallons per minute and 120°F delivered water temperature, a 15 – 17 Btu/hr/ft^2 heat flux or 510 to 600 Btu/hr for a 34 square foot panel was achieved with a room temperature of 70°F.

A pure radiant environment is not likely in residential homes and some air flow can have a beneficial heat transfer effect. Several tests were done on the heating performance of the tube panel with air flow of 100 cfm from the Sanyo mini-split fan operating on low speed. The results showed heating capacity could be increased from 17 to 25 Btu/hr/ft^2 or about 50 percent when compared to the nominal value at 120°F delivered water temperature at 0.3 gallons per minute and a chamber temperature of 68°F.

The Uponor panel sample was tested in the lab for heating and cooling performance as mentioned in Chapter 2. The results are provided in Table 16. These results are consistent with performance information from the manufacturer: 16 Btu/hr/ft^2 for cooling at 58°F supply water temperature and 0.3 gallons per minute and 33 Btu/hr/ft^2 for heating at 120°F supply water temperature at 0.3 gallons per minute.

The conclusions of the radiant heating lab testing task were that the radiant panels had an acceptable heat flux capacity for field testing. The tube panel produced an average heat flux of 16 Btu/hr/ft^2 for the heating season. Increasing the water flow rate or modifying the delivered water temperature both increased system capacity. Higher heating season capacity could be achieved by increasing the hot water flow rate or temperature up to the limit of the water heater capabilities. Using these techniques, the heating season heat flux could be increased to 22 Btu/hr/ft^2. Adding a convective element (100 cfm fan) in a room increased capacity up to 50 percent for heating when starting with nominal conditions. At 16 Btu/hr/sf from the tube panel, the capacity of the heating system is 1.5 times the capacity of the cooling system. At 33 Btu/hr/sf from the Uponor panel, the capacity of the heating system is approximately twice the capacity of the cooling system, satisfying the need identified in Section 3.1.1.

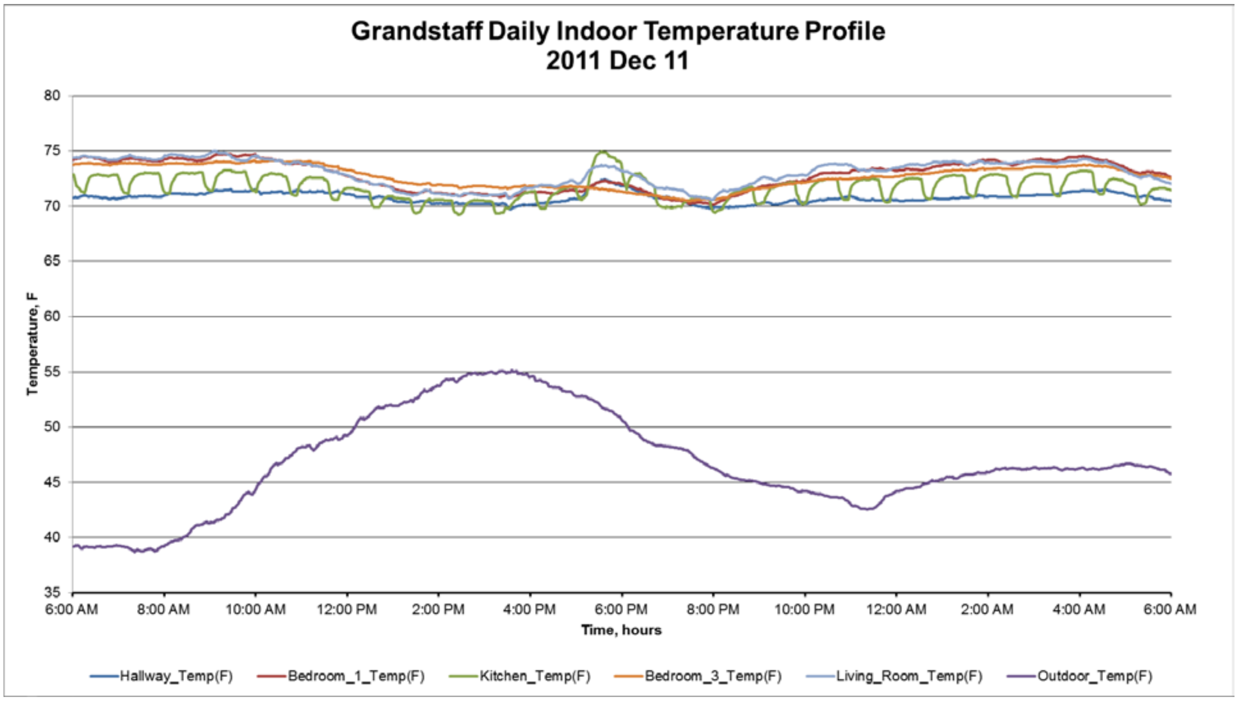
3.3 Field Testing of Radiant Heating Systems

The objectives of the field tests, locations of the sites, installation, and monitoring details were covered in Chapter 2. In this section, the results of the heating field test are provided.

3.3.1 Daily Temperature Patterns and Utility Bill Analysis - Grandstaff

Figure 72, below shows the temperature variation between rooms at Grandstaff on a typical winter day, December 11, 2011. The room temperatures stayed very consistent with each other with about a 4°F variation, and also stayed consistent over time while the system was operating. Note the kitchen temperature variation caused by the cycling of the refrigerator.

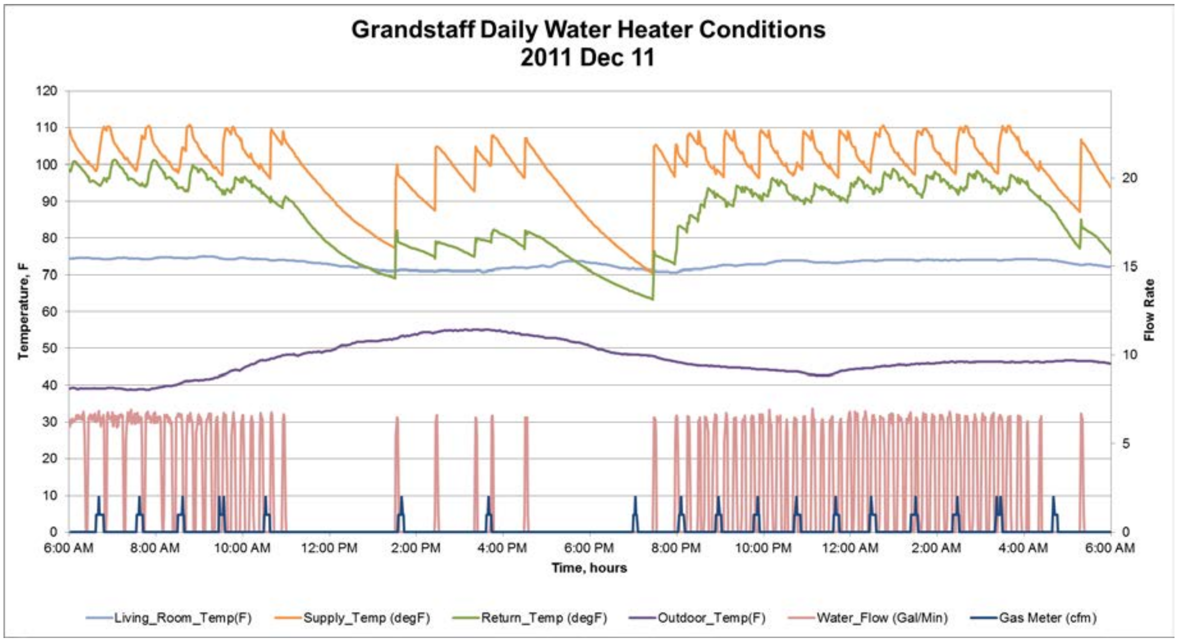
Figure 72: Grandstaff Temperature Profile



Source: Gas Technology Institute

Figure 73 shows equipment cycling over the same test period. The pink line shows the water pump operation caused by regular thermostat cycling at about six cycles per hour until increasing outdoor temperature reduced the demand on the system. The supply water temperature cycles between 100°F and 120°F as the hot water tank thermostat responded to the energy being withdrawn for space heating. The gas meter (dark blue spikes) shows the gas consumption caused by the water heater thermostat cycling and calling for heat. Finally, the living room temperature varied only a few degrees during this process, indicating that comfort in the space was maintained.

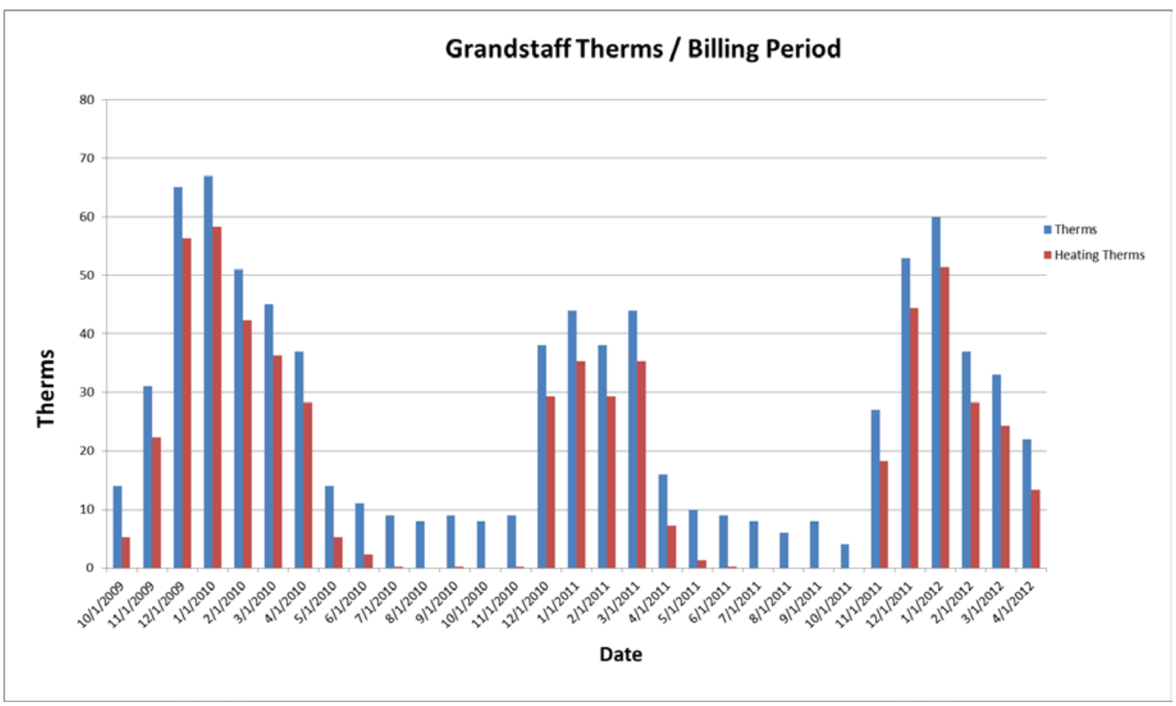
Figure 73: Grandstaff Equipment Cycling



Source: Gas Technology Institute

Figure 74, below shows the utility bill's therms per month for the Grandstaff site over the 2009-2012 billing periods. Heating therms were determined by subtracting the summer monthly average gas usage over the period, in this case 7.8 therms per month.

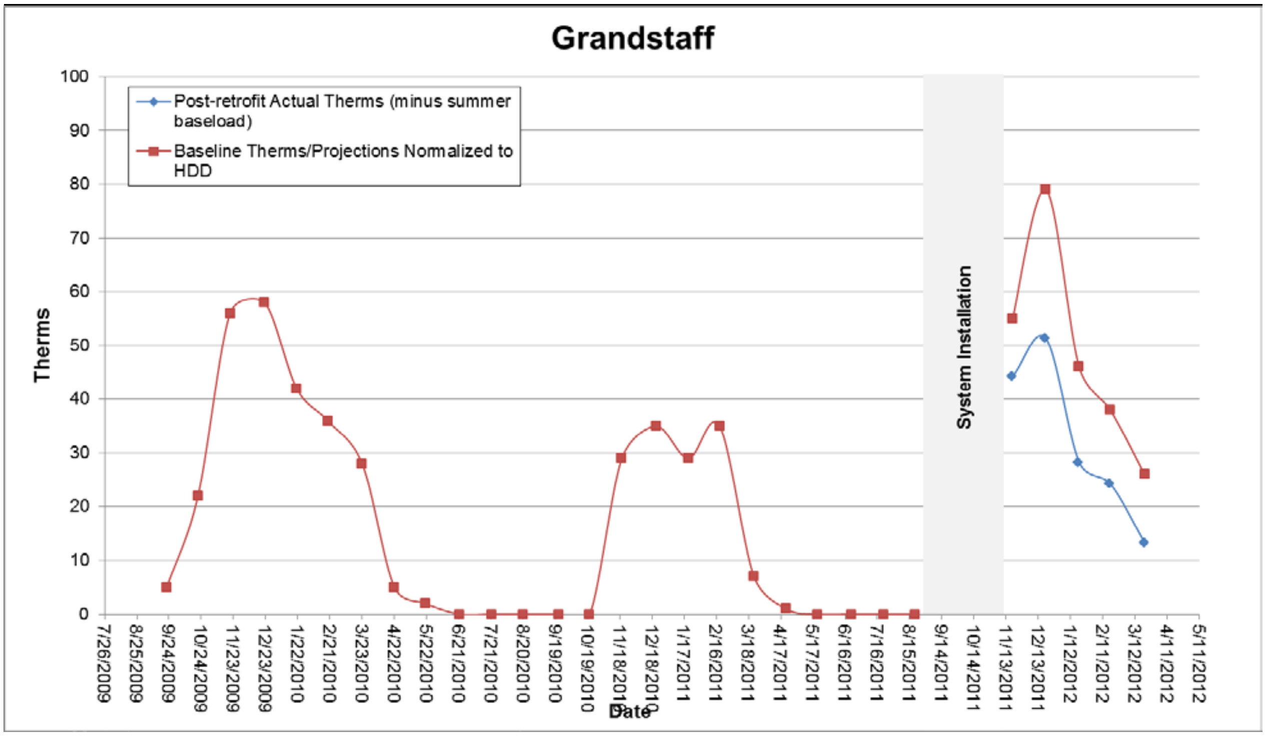
Figure 74: Grandstaff House Therms per Billing Period



Source: Gas Technology Institute

In Figure 75, the monthly heating therms over the same period were plotted and the system installation period was identified. In this figure, the post-retrofit actual therms (blue line) were weather normalized to compare consumption with the 2009-2010 heating season (red line). Note that the consumption patterns for the 2010-2011 heating season showed a significant decline in January that was the result of a long period whenthe house was unoccupied. For this reason the 2009-2010 heating season was chosen for the comparison. The balance point for this house was determined to be 59°F using an r-squared analysis of the energy usage with several balance point options. Heating Degree Days were then determined using the 59°F balance point.

Figure 75: Grandstaff House Monthly Energy Consumption With Savings



Source: Gas Technology Institute

In Table 31, below, the heating degree days for the three periods, the projected therm usage, the actual therm usage and the savings is shown. Note that the heating degree days for the three periods is steadily increasing. Based on this analysis, the weather-adjusted annual savings for the Grandstaff house is 34 percent. If an average of the 2009-2010 and 2010-2011 winters were used as the baseline, the average savings is 16 percent, reducing predicted savings by more than half.

Table 31: Grandstaff House Energy Savings Calculation

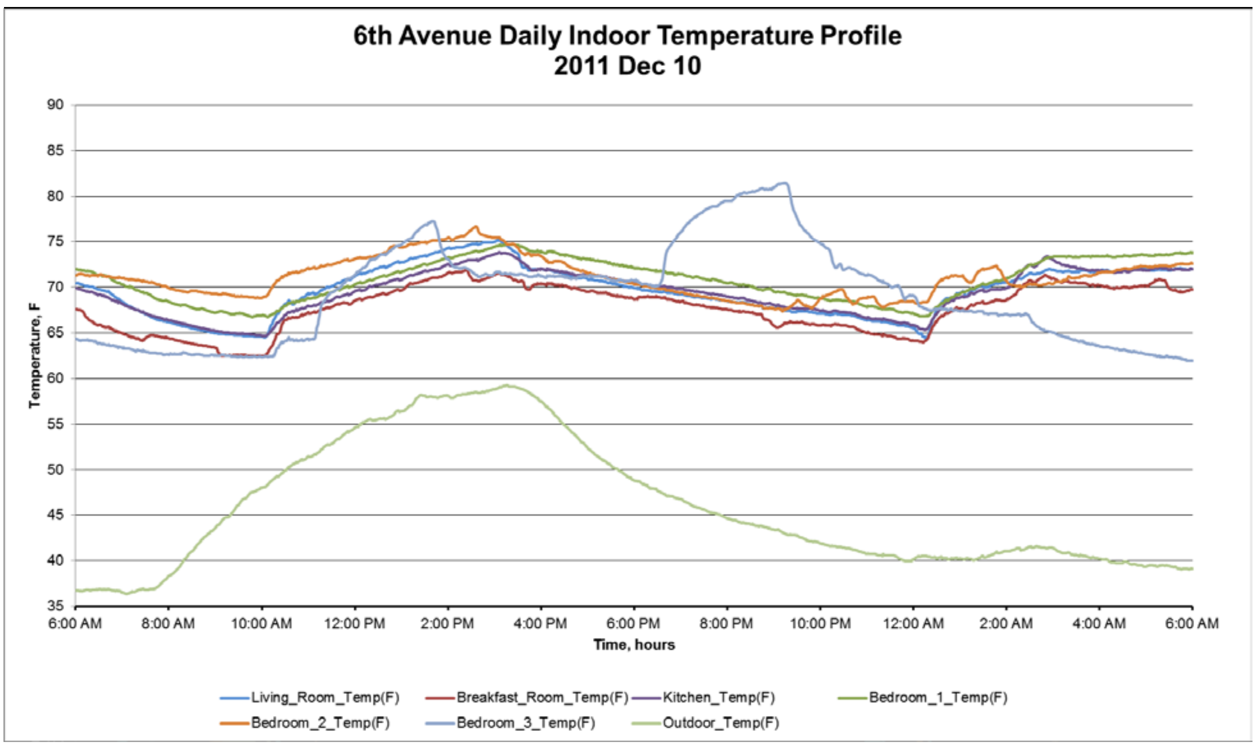
Grandstaff		Actual Space Heat Therms	HDD_59	Projected Therms	Annual Savings	Therms Saved
Pre/Baseline	2009/10 Winter	222	1433			
Pre/Baseline	2010/11 Winter	137	1497			
Post-retrofit	2011/12 Winter	162	1581	244	34%	83

Source: Gas Technology Institute

3.3.2 Daily Temperature Patterns and Utility Bill Analysis – 6th Avenue

A similar analysis was performed for the 6th Avenue house. Figure 76, below shows the temperature variation between rooms at 6th Avenue on a typical winter day, December 10, 2011. The room temperatures were not very consistent with each other with about an 8°F variation, but stayed fairly consistent over time while the system was operating. Note the 3rd bedroom temperature variation - this spike and drop in temperature were likely caused by occupant behavior.

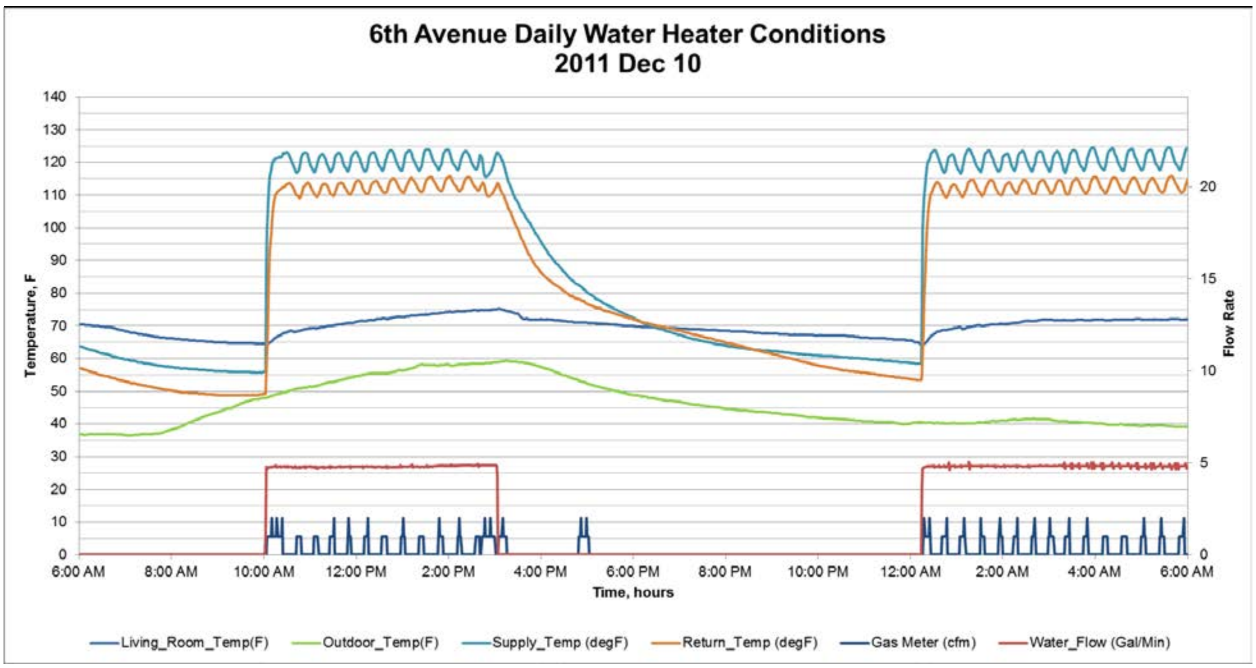
Figure 76: 6th Avenue Temperature Profile



Source: Gas Technology Institute

Figure 77 shows equipment cycling over the same test period. The red line shows the water pump operation. Note that in this case, the thermostat called for heat continuously over a five hour period starting at 10 a.m. and then again at midnight. The thermostat in the water heater reacted to the continuous draw by firing at regular intervals to keep the tank warm. Both the outdoor temperature and living room temperature were rising during the morning period and the outdoor temperature was level while the living room temperature was rising in the period starting at midnight. It appears that the house was responding to a change in the thermostat setpoint during these periods. Since space temperatures were rising, the capacity of the heating system seems adequate for the house. As in the Grandstaff case, the supply water temperature cycles between 110°F and 120°F as the hot water tank thermostat responds to the energy being withdrawn for space heating. The gas meter (dark blue spikes) reads the gas consumption caused by the water heater thermostat cycling and calling for heat.

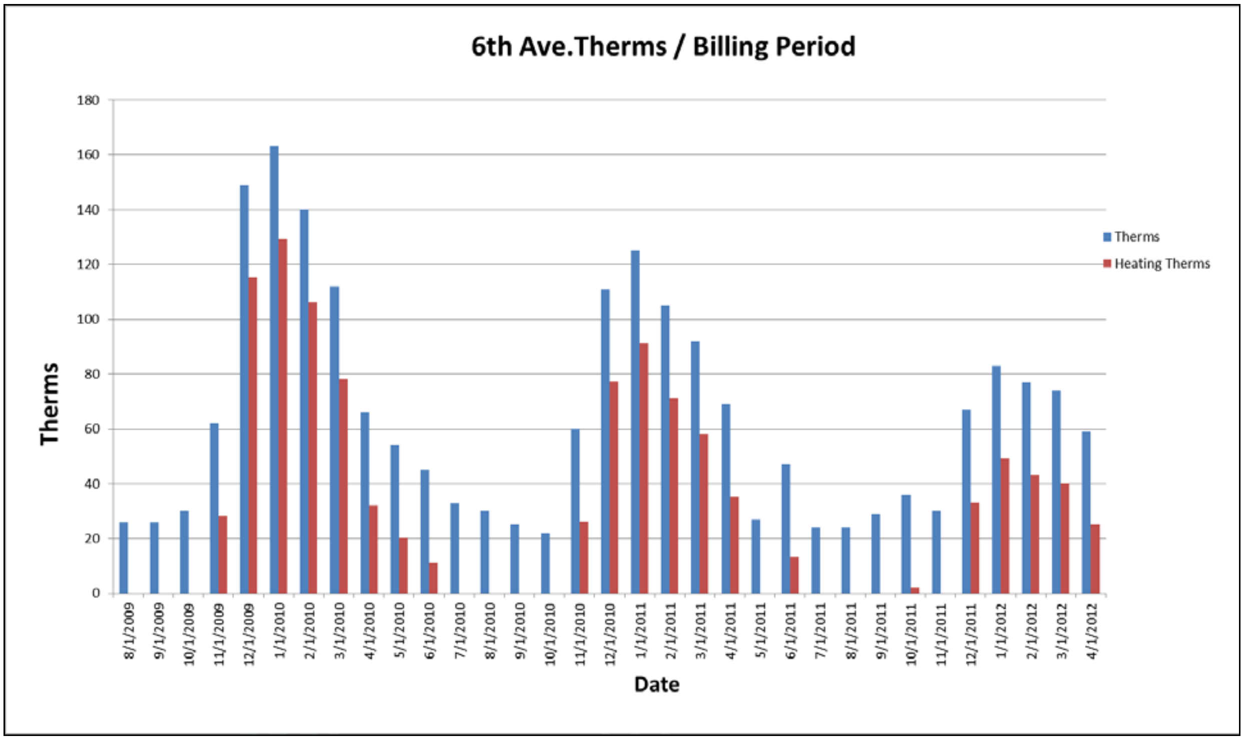
Figure 77: 6th Avenue Equipment Cycling



Source: Gas Technology Institute

Figure 78 shows the utility bill therms per month for the 6th Avenue site over the 2009-2012 billing periods. Heating therms were determined by subtracting the summer monthly average gas usage over the period, which in this case were 33.8 therms per month. The 6th Avenue house had a family of 4, so baseline usage for water heating, cooking, and other uses was higher.

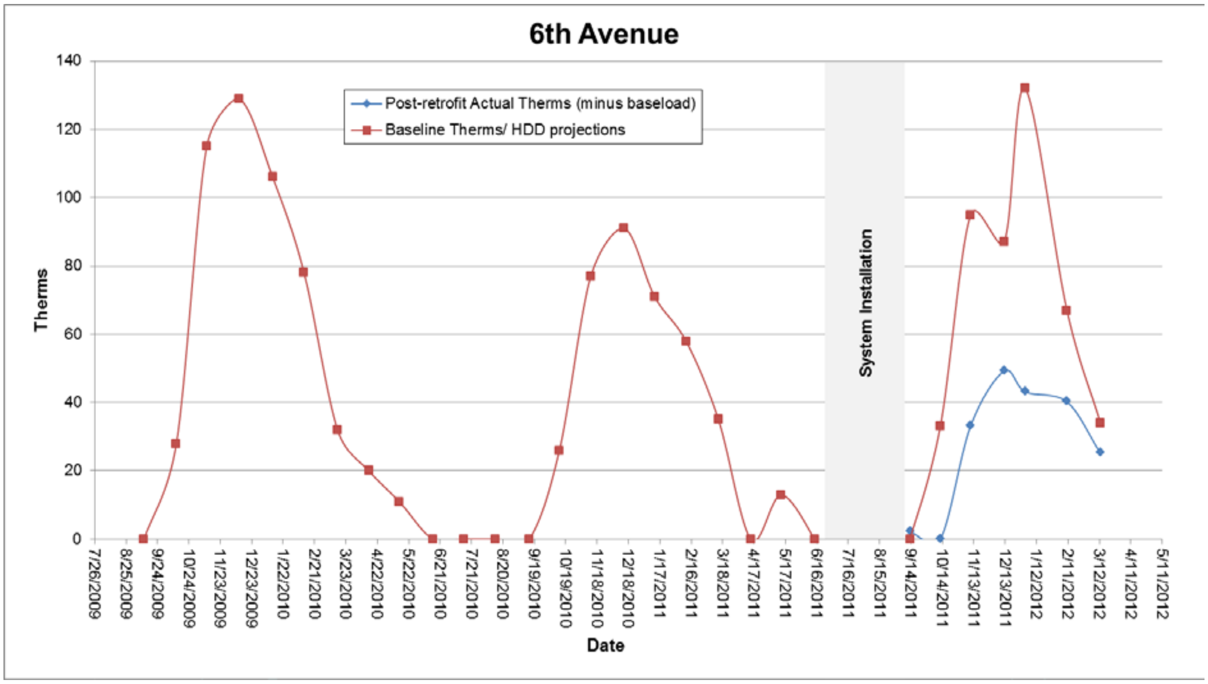
Figure 78: 6th Avenue House Therms per Billing Period



Source: Gas Technology Institute

In Figure 79, the monthly heating therms over the same period were plotted and the system installation period was identified. The post-retrofit actual therms from Figure 79 (blue line) were weather normalized to compare consumption with an average of the 2010-2011 heating season (red line) in addition to the 2009-2010 heating season used in the Grandstaff case, as the unoccupied period in the 2010-2011 heating season was not seen at the 6th Avenue location. Note that the consumption patterns for the 2010-2011 heating season were significantly lower despite having the same heating degree days as the 2009-2010 heating season. The balance point for this house was determined to be 63°F using an r-squared analysis of the energy usage with several balance point options. Heating degree days were then determined using the 63°F balance point. Note that for this house, the balance point was four degrees higher even with more occupants— evidence of the existence of a poorer thermal envelope than Grandstaff.

Figure 79: 6th Avenue House Monthly Energy Consumption with Savings



Source: Gas Technology Institute

In Table 32, seen below, the heating degree days for the three periods, the projected therm usage, the actual therm usage and the savings is shown. Note that the heating degree days had less variation at the 63°F balance point than at the 59°F balance point used for Grandstaff. Based on this analysis, the weather-adjusted annual savings for the 6th Avenue house was 57 percent.

Table 32: 6th Avenue House Energy Savings Calculation

6th Avenue		Actual Space Heating Therms	HDD_63	Projected Therms	Annual Savings	Therms Saved
Pre/Baseline	2009/10 Winter	489	2367			
Pre/Baseline	2010/11 Winter	359	2376			
Post-retrofit	2011/12 Winter	193	2506	448	57%	255

Source: Gas Technology Institute

3.3.3 Field Test Conclusions

The radiant heating system performed very well in the heating season tests. The A. O. Smith Vertex 76,000 Btu/hr input capacity water heater with 96 percent thermal efficiency was sufficient to meet the load of the Grandstaff and the 6th Avenue houses during the winter. The panels performed well in the 18 circuits in the Grandstaff house and 15 circuits in the 6th Avenue house controlling the temperature at night and showing good recovery from setback. There was some evidence of supplemental heating in the 6th Avenue house that resulted in overheating parts of the house above the setpoint. The impact of these behaviors on the capacity of the system could not be determined. The general design conditions of a minimum of 15 Btu/hr/ft^2 of radiant heat from the panels at 120°F and at 0.3 gallons per minute of hot water per panel were confirmed in these studies.

Gas energy savings from Grandstaff was 34 percent compared to its baseline performance, determined by a utility bill analysis. In Grandstaff, the thermostat setpoint was lowered from 70°F to 68°F in the first month of testing by the homeowner due to the improved thermal environment. If the setting would have been left at 70°F for the entire heating season, energy savings from natural gas would have been slightly lower.

In the 6th Avenue house, energy savings from natural gas was 57 percent when compared to the average of the two prior years as a baseline consumption data point. In this house the thermostat setpoint varied significantly at the hands of the occupant, so the assumption was made that the same patterns were used over the three heating seasons. In this house, there was some evidence that the occupant used the oven for space heating. The occupant was warned of the dangers of doing so; however there was some evidence that the practice continued during part of the testing period (a CO detector had been installed). Again, the results were based on the assumption here that the use of the oven for space heating did not change during the three heating seasons used in the study.

The conclusions from the heating season test of the two houses was that the energy reduction associated with the radiant heating system and the measured home performance improvements produced an average savings of 45 percent for the two houses in the Sacramento area with improved comfort. It was not possible to separate the effects of the two factors in this study, however a predicted savings for the increase in efficiency of the water heater alone would yielded a 15 percent savings for Grandstaff (vs. 80 percent AFUE furnace) and 30 percent savings for 6th Avenue (vs. 65 percent efficient heating system), leaving approximately 25 percent savings to be spread between thermal envelope improvements, the performance of the radiant heating system and the use of a lower thermostat setpoint.

Refer to Appendix A for an analysis on ideal locations in Southern California for the radiant heating and cooling system.

3.3.4 Field Test Issues Identified

The field test provided a good venue for identifying possible improvements in the system, as indicated in each of the subsections above. For both the heating and cooling system design and operation, problem areas identified for further investigation include:

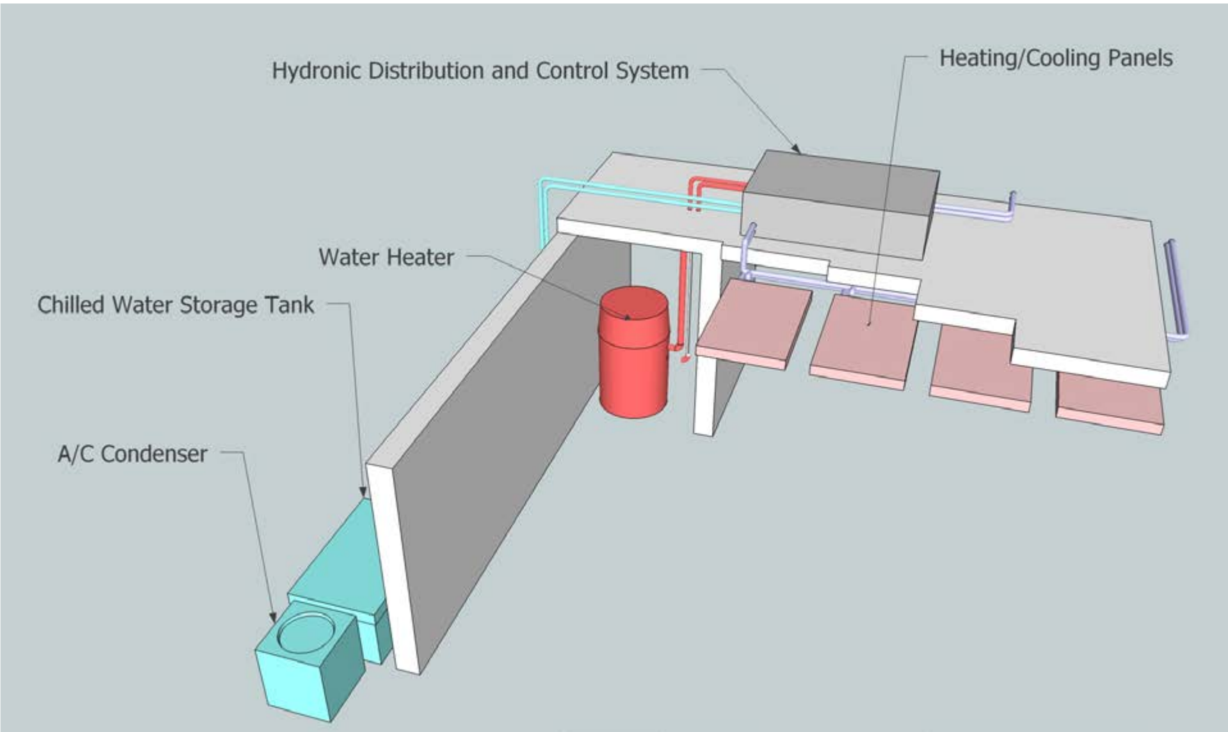
1. The chilled water storage tank required stirring to avoid water freezing up on the surface of the DX (direct expansion) coil. A small water pump and a propeller-type stirrer were both evaluated in the project; higher flow rates worked better. This could be eliminated by increasing the evaporator temperature or redesigning the evaporator.
2. The three-way valve providing changeover from heating to cooling was operated by a relay energizing a motor. Three-way valves with integrated operation are available for commercial and industrial uses, but the price is not viable for residential applications.

3. The panels require many tubing connectors, most of them located in the attic. A leak requires accessing the attic and digging through insulation to locate the failed connector. The only fitting leak in the project was just above a light fixture in the bedroom, and it was repaired before additional installation was installed. A better design would be to have all tubing connectors accessible from below the ceiling level, although the drywall would have to be cut and repaired.
4. The metric size of the Uponor panel required framing below the ceiling to “convert” from metric to standard U.S. framing. If the product is sold in the U.S., it would have to be modified to fit standard framing.
5. Behavioral issues are difficult to control in the field. Close monitoring and a lock-out thermostat should be considered.

CHAPTER 4:
Market Paths

The first step in the development of the system is to perform a high-level economic assessment of the technology in California climates as a precursor to developing market paths. A basic system, consisting of the components in Figure 80, below, was designed and a cost was determined. A detailed building energy analysis was conducted on two homes in seven climate zones to determine the economic potential and payback period.

Figure 80: System Schematic



Source: Gas Technology Institute

A one-story 1764 square foot house and a two-story 2312 square foot house were modeled in California climate zones two, eight, nine, 10, 12, 13, & 15 using the Energy Commission approved program MICROPAS. These zones were chosen as representative of California's major cooling climates. The performance of a properly-sized standard forced-air cooling system was modeled based on hourly simulated loads and temperatures generated in MICROPAS. Energy consumption for each hour of the year was obtained and later utilized to calculate the monthly and yearly cost of cooling using a tiered rate structure. Climate zones two and 13 were assigned to PG&E, zones eight and nine to SCE, 12 to SMUD, and 10 and 15 to SDGE (Sempra). These same temperatures and conditions were then used to model a radiant cooling system with off-peak storage. Similarly, the cost of cooling was estimated, only now applying a time-of-use rate structure. Savings over the forced-air system were demonstrated while reducing on-peak cooling demand to near-zero (as determined from the 120 watt water pump). The initial cost of the radiant cooling system was compared to the cost of forced air systems for different incentive levels. On the basis of this first cost and operating cost analysis, the payback period for the radiant heating system was determined in the following sections.

4.1 Market size

Table 33 shows the total number of single-family homes in the targeted climate zones.

Table 33: Total Housing Stock by Climate Zone

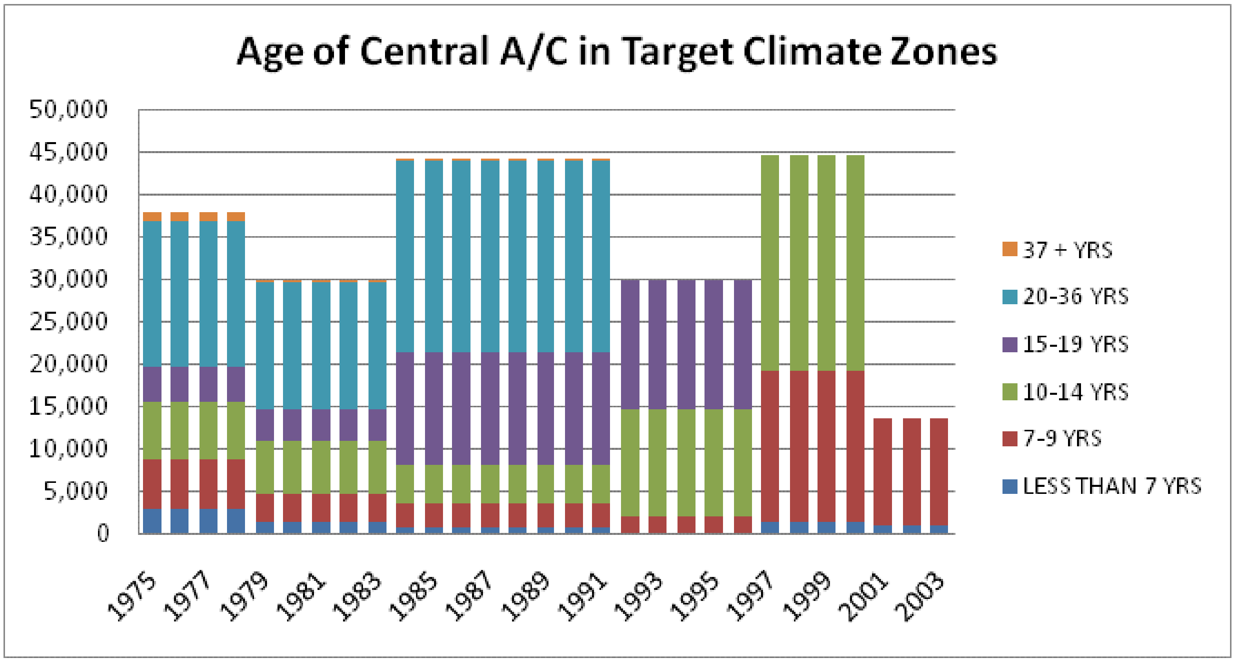
Representative City	Climate Zone	Number of Houses
Santa Rosa	2	170,892
El Toro	8	485,342
Pasadena	9	632,361
Riverside	10	597,482
Sacramento	12	572,501
Fresno	13	321,981
El Centro	15	49,777
Total		2,830,336

Source: Western Cooling Efficiency Center

The most promising application of this cooling system is in new single-family construction, because the implementation of radiant cooling and thermal storage during construction would require minimal extra construction work.

In addition to new homes, there is an expected market in retrofit applications, as people add or replace forced air systems. Figure 81 shows the current ages of central air-conditioning systems in the relevant zones sorted by house age. With a 15 year expected lifespan, it is clear that, while many houses have had their air conditioning replaced, a large number of the older systems are overdue for replacement. From this data the team calculates that there are already some 500,000 systems aged 15 years or more, with approximately 50,000 additional systems reaching that age every year. Even before the recovery in the rate of new builds is factored in, the combined new build and retrofit markets amount to some 70,000 systems per year. At a typical cost for a standard HVAC system for a single family home of approximately $7500(Radcliff) this amounts to over $500 million per year.

Figure 81: Age of Central Air-Conditioning Systems in Houses in the Target Climate Zones Sorted By House Age



Source: Western Cooling Efficiency Center

4.2 Market Barriers

New technologies for residential cooling face a number of impediments. The most significant is the near monopoly of residential cooling by compressor driven systems. Most available estimates show vapor compression systems as having over 95 percent of residential market share. There are a number of reasons generally cited for this (Feustel, 1991):

- The first cost of vapor compression systems is comparatively low
- Equipment, parts and servicing are readily available
- Systems are mechanically reliable
- Availability of a wide range of capacities to satisfy any building size
- Controls are relatively simple and response times are short

This market domination is to an extent self-perpetuating: consumers are faced with the choice between a technology that is seen as convenient, reliable, well established and relatively inexpensive and a number of alternatives which are effectively seen as lacking these attributes.

Radiant cooling faces an additional barrier, not seen by evaporative coolers (which, like vapor compression systems, deliver cooling via a chilled airstream) in that the method of cooling is unfamiliar to most consumers who will, in general, have had experience of vapor compression systems.

The barriers to market entry can be addressed individually to determine possible strategies for facilitating the adoption of radiant technologies.

4.2.1 Consumer Reluctance

Consumer reluctance to adopt new technologies can be broadly divided into four categories (Tesink, 2005): tradition, ease of use, image, and cost.

4.2.1.1 Tradition

This is essentially an inertial barrier - "Forced air systems worked for my parents and they work for all my friends and neighbors – why would I not use one?" For a technology as well embedded as forced air air-conditioning, this barrier is unlikely to be rapidly broken down except when dealing with innovators/early-adopters that are by definition not bound by tradition. When combined with the risk aversion typical of most consumers, this means that the radiant system needs to demonstrate substantial advantages over the forced air system before it becomes a likely choice. These possible advantages can be divided into usability issues and cost issues, which address two of the other categories of reluctance.

4.2.1.2 Ease of use

Radiant systems operate in a fundamentally different way from forced air systems. The most immediately noticeable difference is likely to be the pull down time, which will be longer for the radiant system. This will possibly be seen as a problem by users used to forced air systems, and will necessitate a behavioral change by the user which will rely on the appropriate use of setbacks and programmable thermostats. The issues surrounding the use of programmable thermostats have been extensively researched (Peffer, 2011). Much of the research suggests that the majority of customers do not use these thermostats correctly, so this is a problem for forced air systems as well as for radiant systems.

Once comfort issues are factored in, the radiant systems are seen to offer many benefits with respect to forced air systems:

Table 34: Benefits of Radiant Systems

Thermal comfort	A well-designed radiant system will deliver greater thermal comfort than a forced air system due to the better balance of radiative and convective heat transfer between the room and the occupants. This allows the thermostat settings to be raised (in summer) or lowered (in winter), thereby providing a cost saving, without sacrificing comfort.
Noise	Radiant systems are virtually silent, with the water pump usually being located in a garage or mechanical room, far from occupied spaces
Visibility	The absence of supply or return grills results in a system that is to all intents invisible
Air quality	One of the most common complaints about forced air systems concerns their tendency to distribute dust, odors and germs throughout a house. In contrast to whole house air movement, a radiant system creates very gentle room air circulation[1]

Source: Western Cooling Efficiency Center

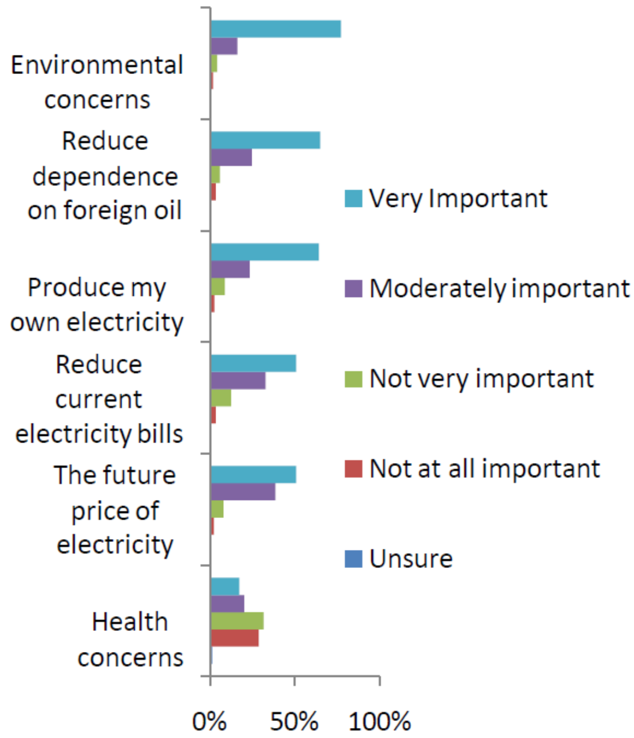
Any or all of these could be the selling point for a given customer.

4.2.1.3 Image

Being largely invisible within the home, air conditioning is not a particularly glamorous technology. It is thus unlikely that radiant cooling systems can be marketed on the basis of image. Lessons can be learned from the early years of hybrid vehicles – some studies (Heffner, 2011) have shown that many purchases were driven by environmental concerns. Similar results were found for photovoltaic systems – a 2009 survey by the Solar Electric Power Association, SEPA Report # 06-09, found that environmental concerns were the most important motivation when deciding to purchase photovoltaic systems (Figure 82).

[1] Make up air can be provided by a separate dedicated outdoor air system, which will circulate substantially less air than a forced-air AC system

Figure 82: Purchasing Motivations for PV Systems



Source: SEPA Report # 06-09

4.2.1.4 Cost

The cost comparison between forced air and radiant systems can be divided into first cost and cost of use.

First costs: on the basis that the radiant system will use an evaporator of similar cost to the forced air system, and a similar compressor, the price difference for materials will be due to the difference in cost between the two delivery systems: radiant (surface, tubing, and pump) and forced air (registers, ducts, and blower), with the storage tank appearing as an additional cost for the radiant system. Installation costs for the radiant system will be dependent on the actual type of radiant surface used. Labor costs will differ for installation of the two radiant surfaces studied in this project. The below ceiling retrofit panels offer lower installation cost as there is no finishing required, giving an anticipated marginal cost for installation by a trained installer of less than $1 per square foot.

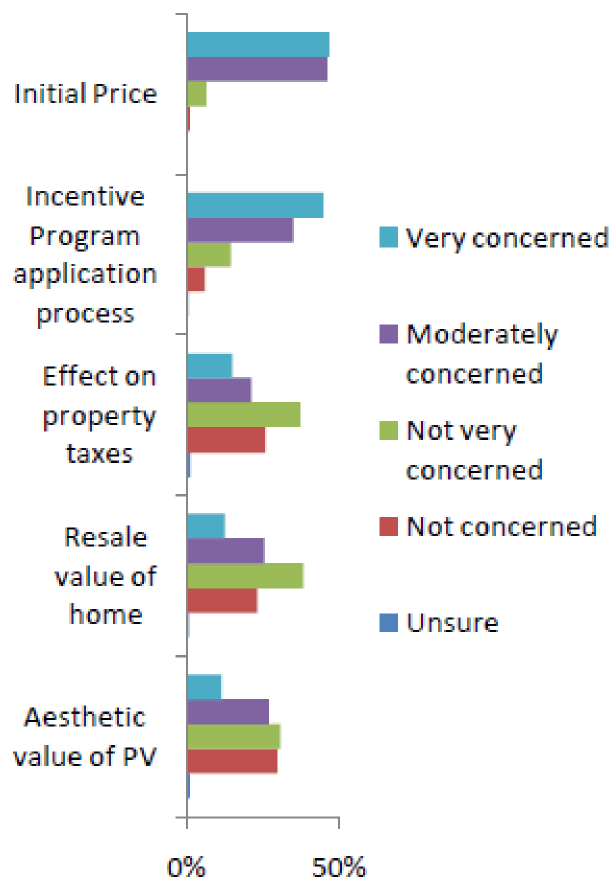
The drywall based panel from Uponor designed for new build or deep retrofit (bare studs) installation has a higher anticipated cost due to the need to mud and tape the panels. This cost may be partially offset by the fact that the radiant panel replaces the drywall that would otherwise be installed, thereby saving both material and labor costs (note that this saving only applies if the system is installed as part of a project that would require a new or replacement ceiling). The anticipated installation cost is in the region of $0.50 - $3.00 per square foot.

The drywall based panels supplied by Uponor for this project have a material cost (at retail prices) in the region of $1.50 per square foot. The retrofit panels have a material cost close to $5 per square foot at volume prices. There is room for substantial lowering of this cost, but this would require an investment in tooling.

Combining the material and installation costs gives a range of $2 to $6 per square foot for the two systems. As a comparison, the HVAC system for a new 2500 square foot home costs approximately $1 per square foot (Radcliff 2011). Radiant surfaces in the climate zones considered in this project will need to cover at least one third of the ceiling area of a well-built house, or approximately 830 square foot, leading to a target price in the region of $2 per square foot of panel area in order to compete with the cost of forced air systems.

From the study of photovoltaic systems referenced above (SEPA Report # 06-09), when purely financial considerations are studied, the initial cost is most important, followed by the availability of rebates (Figure 83).

Figure 83: Purchasing Motivations – Financial



Source: Solar Electric Power Incentives

The savings associated with radiant systems are dependent on what comparison is being made. It is tempting to compare a hydronic radiant system with a typical forced air system and make the case that the radiant system is far more efficient due to the elimination, primarily, of losses due to duct leakage, which will typically be in excess of 25 percent. This is, however, not a robust argument for two reasons. First, it is obviously not comparing like with like – a well installed radiant system should self-evidently be compared to a well installed forced air system; and secondly, as building codes become more strict and better enforced, forced air systems will tend to be better installed.

On this basis it is clear that in order to develop the market for radiant cooling systems it is necessary to market them as greener alternatives to forced air systems, to develop rebate programs with utilities, and to emphasize the added comfort provided by radiant systems.

4.3 Innovation Cycle

New technologies enter the market according to a well-established pattern: purchasers are described as innovators, early adopters, early majority, late majority and laggards. Residential radiant cooling systems are currently very early in the cycle, being purchased only by a small number of innovators and installed by specialized independent contractors. The challenge for radiant cooling at this point is to move to the point where it will be taken up by early adopters, who are more sensitive to pricing.

4.4 Stakeholders Analysis

4.4.1 Manufacturer Perspective

Because the radiant system consists of three readily separable components (radiant surface, chiller, and storage) there is an opportunity for innovation on numerous fronts.

4.4.1.1 Radiant surfaces

There are a number of manufacturers of radiant panels designed for commercial use, which are mostly aluminum with soldered copper tubing. These are primarily designed for use with drop ceilings, which when combined with their relatively high cost in a comparatively mature market (typically $12-$15 per square foot) suggests that there is little opportunity for price reduction starting from these designs. There are a number of smaller companies (e.g. Talbott Radiant, Messana) offering more innovative products where there is a higher likelihood of price reductions. These companies are already offering products at a price point below $15 per square foot and anticipate substantial price reductions in the future due to economies of scale (Marchesi).

4.4.1.2 Chilled water storage

Chilled water storage tanks in sizes suitable for this application (in the region of 500-1000 gallons) are rare. Most small insulated tanks are designed for solar thermal water heating and are 200 gallons or less. The tanks used for this project were custom built by Integrated Comfort Inc. and as such the cost included development cost. Anticipated bulk costs for the tanks (in quantities of 1000) are in the region of $3.50 per gallon (Bourne).

In addition to the hard tanks used in the field tests, a soft sided tank was developed for the project in conjunction with American Solar Technics. The cost for this tank would be substantially lower, in the region of $2 per gallon, but the current design is only suitable for indoor use. It remains a viable possibility for homes with a suitable indoor space.

4.4.2 Contractor Perspective

Essentially, contractors will sell and install any system that they feel will prove profitable. This involves a risk/reward trade when it comes to new technologies, as there is likely to be an unknown liability issue.

4.4.2.1 Liability

This can be resolved into two principle components: the possibility of the system failing to cool the building adequately and provide occupant comfort; and system failures leading to property damage.

The first issue can be dealt with during the design phase. Tools for modeling building loads are sufficiently well developed and robust as to all but eliminate this issue. The combination of a radiant cooling system with a dedicated outdoor air system has been extensively studied (Conroy, 2001) and has been shown to be more than capable of meeting cooling and ventilation demands. The issue of condensation, and consequently the possibility of mold formation, has been a significant factor in delaying the adoption of radiant cooling when compared to radiant heating as mold formation impacts both the comfort and damage aspects of liability concerns (Radcliff). This is again a point that can be adequately addressed during the design phase.

Because of this, the main liability concern will be leaks. Current radiant panels use either copper tubing or—as in this project—PEX tubing, both of which are widely accepted for hydronic applications, and are code complaint in most areas. Leaks are only likely to occur at interconnects, so the design of the radiant surface and the piping layout within it are of primary concern. The two panel designs used in this project both result in a substantial number of interconnects in the attic space of the test houses, but other possible designs for radiant ceilings (e.g. the X-lath design used by Talbott Radiant) have none, with large loops of PEX tubing connecting back to a central manifold, which is readily accessible. Building codes now allow the use of PEX fittings in walls, where the consequences of leaks would be similar to ceiling leaks, yet plumbers are happy to plumb houses with PEX.

4.4.2.2 Profitability

There is currently a large price premium to install a radiant system, hence the low market penetration and the restriction to innovators. In order for prices to come down while systems remain profitable to install, either the cost of components or the cost of installation (or both) will need to fall. Currently, unlike forced air systems, radiant systems are effectively all custom installations. Until the cost reductions associated with larger scale production and refined installation methods reduce the price, it is likely that radiant systems will need incentives to drive their uptake. A shift in approach from contractors may be necessary: to quote Robert Bean on hydronic systems in general "I can't see this industry growing without contractors getting over the fact that hydronics is a consumer product. Treating these systems as customizable designs is destroying the image of hydronics. We've been fighting this for 30 years, and we haven't gotten anywhere because we still define this as custom work."(ACHR)

4.4.3 Utility Perspective

The role of utilities in the adoption of radiant cooling systems will primarily be to provide rebates and incentives. These incentives will have a significant impact on the marginal cost of choosing a radiant system. Generally, utility incentives for residential technologies are in the form of a lump sum payment, rather than the model for commercial technologies of payments based on kWh saved and kW of load shaved.

4.4.3.1 Peak power reduction

The use of thermal storage is the main driver of peak power reduction, as it eliminates all peak load except the circulator pump and control electronics. It is here that utility incentives are most likely to have a significant impact on uptake. It is also here that the argument can most easily be made as the removal of peak load is easier to quantify than the efficiency savings. Time of use pricing, along with being an incentive for installation of load shifting technologies, additionally acts as a motivation for customers to install thermal storage systems.

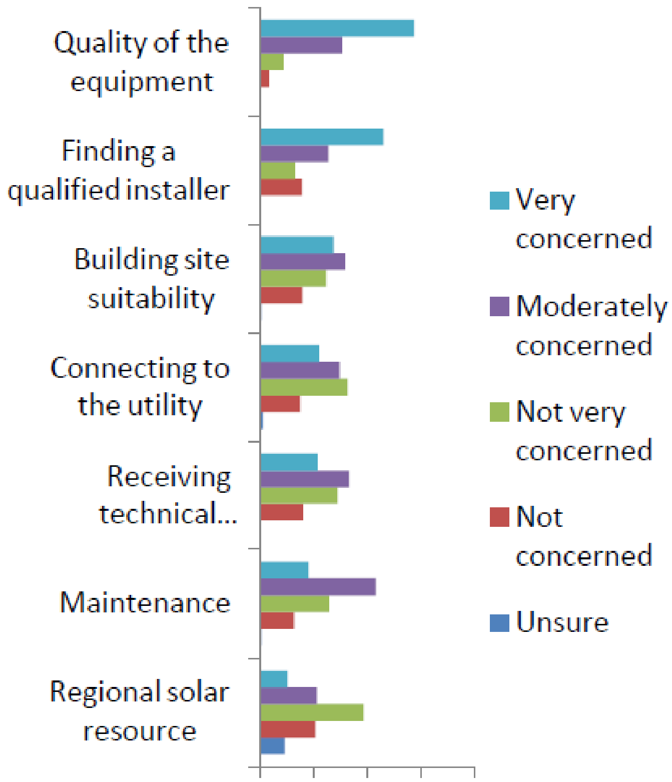
Rebates for residential thermal storage systems are not unprecedented, but can show the risks of prematurely marketing technology. A pilot program developed by SMUD in the 1990s offered a rebate of 5.72 cents per kWh for customers installing approved thermal storage systems. The company responsible for installing the systems is no longer in business – the resultant loss of support for maintenance has led to all the systems being decommissioned and the program being closed. Reliability is likely to be a significant driver of market uptake (Figure 84). This suggests that systems at this stage should be developed such that maintenance and repairs can be carried out by contractors without specialist knowledge of the design, requiring only standard plumbing and HVAC skills.

4.5 Overcoming Barriers to Entry

The benefits of radiant cooling with thermal storage are potentially significant in terms of peak load reduction and energy savings, and are therefore amenable to utility incentives. The penetration of radiant heating into the marketplace demonstrates that the public is becoming aware of, and increasingly open to, the concept of radiant heat transfer. Specialist contractors are reporting steady business installing radiant systems and interest from major homebuilders (Talbott). Based on discussions with these firms, the two most significant barriers to overcome appear to be customer's lack of knowledge, and cost. Successful demonstration projects, custom installations, and dissemination of results will begin to inform the public and promote further development of radiant surfaces, storage tanks, and installation methods will begin to reduce the costs.

This project has focused on successfully demonstrating the technical feasibility of simple residential radiant heating/cooling systems without the need for dedicated outdoor air systems or complex controls. The next steps to be taken will involve establishing one or more pilot programs to further demonstrate the reliability of the system and capture data on the savings available.

Figure 84: Purchasing Motivations – Technical



Source: Solar Electric Power Incentives

CHAPTER 5: Measured Home Performance and Radiant Heating and Cooling Systems

5.1 Overview

The current design and installation practices for insulation, air sealing and HVAC systems, whether conventional or radiant, are fragmented. Typically, in both new construction and retrofit markets, these three functions are performed by different subcontractors. From a commercial perspective, the results are acceptable because they provide an apparently low-cost result. The bidders who can comply with the primary measurement criteria of lowest cost and quickest installation which meets code requirements succeed.

From an energy perspective, the results are extremely poor. Current codes do not require measured energy performance. So the consumer has no means to objectively judge the value of air sealing, insulation and HVAC systems other than their separate installed costs. So the small-but-critical differences in design and installation which improve energy performance are not rewarded by commercial success. However, when these elements are measured and integrated, the energy savings are substantial and the installed cost can be equal to the poorly-performing, non-integrated current practices.

For example, at the same site in Redding, CA, two identical houses were erected in a development of high-end, energy-efficient homes. Installed costs were similar for both homes. The first house used conventional design and installation practices, but used advanced technology heating and cooling equipment (a geothermal heat pump). The second only used conventional heating and cooling equipment, but the home used integrated design and installation with measured home performance techniques. That home was able to maintain comfort using a system with less than 30 percent of the cooling capacity of non-integrated equipment sizing assumptions. Its measured annual energy consumption was 60 percent less than the home which used non-integrated design and installation, in spite of the theoretical energy advantage of that home's geothermal heat pump (Springer, 2006).

Measured home performance (MHP) contracting is different from other approaches to saving energy and competitively entering the home improvements market. The principal difference being that measured home performance contracting is an in-process, worker measured and tailored-to-fit approach to a project which provides quantifiable results as opposed to estimated best-case scenarios. Measured home performance—also known as Advanced Integrated Installation Methods, Home Performance, Building Performance, House-as-a-System, Systems Approach, Energy Efficiency, Efficiency First, and Reduce Before You Produce–is the direction of the future. This approach requires that two new concepts be implemented as buildings are constructed and renovated:

- Assure that all of the energy features are well-designed, installed to perform at their rated efficiency, and complement each other. For example, reducing the thermal envelope losses reduces the size of the heating system which in turn has an impact on the location of the diffusers in the rooms, which impacts the design of the duct system.

- Confirm the installed performance of all of the building energy features, using building performance test equipment, in order to assure that the specified installed performance levels are being met. For example, the desired infiltration level should be measured by a blower door before the contractor leaves the site. Similarly the air performance of the HVAC system should be measured before the HVAC technician leaves the site.

Installed performance of the energy features, when tested, can be only about half what is expected. For example: In a California Energy Commission funded research project (Proctor, 2011), 10 homes were performance tested and then quick (2 technicians for one day) retrofit repairs were made. The repairs only focused on system refrigerant charge and system airflow – and ignored all other opportunity for improvement. The increase in delivered performance averaged 26 percent. To install the system originally with performance in mind, in all categories, would yield much greater system improvement – case studies typically show a doubling the system performance.

A synergy exists between building energy features. A well-insulated home, with low air leakage rates and good windows will complement the heating, ventilation, and air conditioning (HVAC) system. Yet individuals too often see HVAC contractors replacing systems, in homes with no insulation, high air leakage rates, and old single glazed windows.

The benefit of pursuing the integrated approach and capturing the potential synergy is complicated and can best be described with an example. Case studies and demonstration projects typically show that a good thermal enclosure only has, on average, half of the heat loss and heat gain of a typical thermal enclosure. A properly installed and properly performing HVAC system delivers, on average, twice the performance of a typical HVAC system. Halving the envelope loads and doubling the system performance decreases the needed HVAC equipment to one-quarter of what would typically be required. With the performance increase and load decrease the HVAC equipment needed would be small enough to fit inside the pressure and thermal boundaries – further decreasing the loads. Other examples of the integrated approach:

- An architect, designing space for all the mechanical equipment inside the pressure and thermal boundaries.
- An insulation installer burying ducts in the loose fill attic insulation.
- An insulation installer locating the thermal boundary at the roof assembly which brings the mechanical equipment located in the attic inside the pressure and thermal boundaries.

Conceptually, implementing MHP should not be difficult. For the HVAC subcontractor to confirm the performance of his work it would mean; (1) measuring the air flow at each supply grille, (2) measuring the temperature at each supply grille and the return grille, and (3) calculating the heating energy or the sensible cooling energy delivered, and comparing the delivered Btu's to the equipment manufacturers' specification at those conditions. If the delivered Btu's do not match the manufacturers' data, repairs would need to be performed. These HVAC system measurements will require about one hour and be performed by one of the technicians that is already on site, making the test cost about $50.

For the insulation subcontractor there are only two steps to confirm the performance of his work; (1) perform a single-point blower door test, to confirm a tight envelope, and (2) film the structure using infrared thermography to confirm no thermal defects in the opaque assemblies. These two quality control tasks will take about thirty minutes and cost about $20, since they are performed by an installer that is already on site.

The hardest part is to confirm that the energy features complement each other. This requires coordination between several people that typically never talk to each other. The HVAC subcontractor must understand and rely on the performance of the work done by insulation subcontractor as the system sizing depends on the performance of the thermal enclosure. The performance of the insulation subcontractor's work is dependent on the quality of the framing and the proper installation of draft stops during original construction.

Though conceptual implementation seems difficult, it is doable. Obstacles occur at many levels of the process, but they are clear, understandable and can be overcome with effort.

There are three major barriers to implementing measured home performance and capturing the well quantified opportunity:

1. Homeowner perceptions
2. Equipment marketing
3. Misaligned interests

The typical homeowner perceives that the only difference from one energy feature contractor to the next is price. Work performed by one licensed and bonded energy feature contractor will be the same as any other, since everyone is working to the same codes and standards, and all work is inspected by the local building department to assure compliance with the codes and standards. Contractors reinforce this perception by focusing heavily on price. Too often homeowners hear from contractors; "I will beat any price," "My work will pay for itself in three years," "I will install more insulation than the other contractor for the same price," "I will install a five-ton air conditioner for the same price that the other contractor gave you for a three-ton air conditioner."

Performance is not often considered until equipment efficiency is discussed. The performance specifications then alter the system or product price – but the "installed performance" may not match the "performance specifications" due to the installation quality. Paying for expensive high performance equipment or high performance products that perform at half of their specified performance level is the standard rather than the exception.

The relationship between the installing contractor and the homeowner are not always aligned. It is in the homeowner's best interest to have the most cost effective balance of; equipment efficiency, system installation quality, ease of equipment maintenance, equipment and system durability, delivered comfort levels, and the proper selection of energy features that provide synergy. Since the interests of the homeowner and the installing contractor are not aligned there is an adversarial relationship rather than one where two entities, contractor and homeowner, are working together to provide the best solution.

Interests could easily be aligned if the contractor utilized MHP since the tested high-performance installation and proper energy feature selection provide utility bill reductions large enough to cover the loan payment needed to pay for the work.

Other barriers to MHP include:

- Utilizing computer modeling to determine the performance of the new energy features. The few case studies and research efforts possessed have consistently showed that when MHP is used, results exceed the modeled performance by 30 percent to 60 percent. When MHP isn't used performance results vary.
- Utility incentive programs that use computer modeling to predict energy savings, but do not look at the actual savings lead to industry standard work rather than MHP.

The most important question to ask is whether these barriers must be overcome and whether MHP should become the industry standard. Implementing MHP avoids several of the pitfalls associated with current programs and practices:

- Ratcheting up the energy efficiency specifications of buildings.
- Focusing heavily on equipment efficiency, rather than installed performance.
- Rely on green programs for above code minimum energy performance.
- Adopting industry created best practices standards.
- Paying for third party verifiers to ensure the energy features perform.

California has been on this path for more than 30 years and in the most recent Energy Commission funded research tremendous opportunity for improvement was seen, an improvement that MHP would capture.

5.2 Approach

In this project, applying MHP to the field test sites was a critical success factor: reducing the cooling load reduced the volume of chilled water storage required to shift the cooling load 100 percent to off-peak hours. The physics is simple. If 1000 gallons of storage is needed to cover a 12-hour 4 ton cooling load, then 500 gallons of storage could shift a 12-hour two ton cooling load to off-peak. In addition, the techniques practiced in measured home performance had been taught by a few skilled practitioners, but had not yet been written down and communicated to the larger contractor community. In this project, the team created the *Measured Home Performance Guide to Best Practices for Home Energy Retrofits in California*, developed and conducted training classes, and applied those techniques to the field test sites. In this chapter, the published guide is summarized and the application of the techniques to the field test site is covered. Chapter 6 covers the technology transfer effort in detail.

5.2.1 Development of the Measured Home Performance Best Practices Guide

Figure 85 shows the front cover of the *Measured Home Performance Guide to Best Practices for Home Energy Retrofits* (Chitwood and Harriman, 2012) that was produced in this project. The guide is available from amazon.com and online for free download at http://www.measuredhomeperformance.com.

Figure 85: Measured Home Performance Book Cover

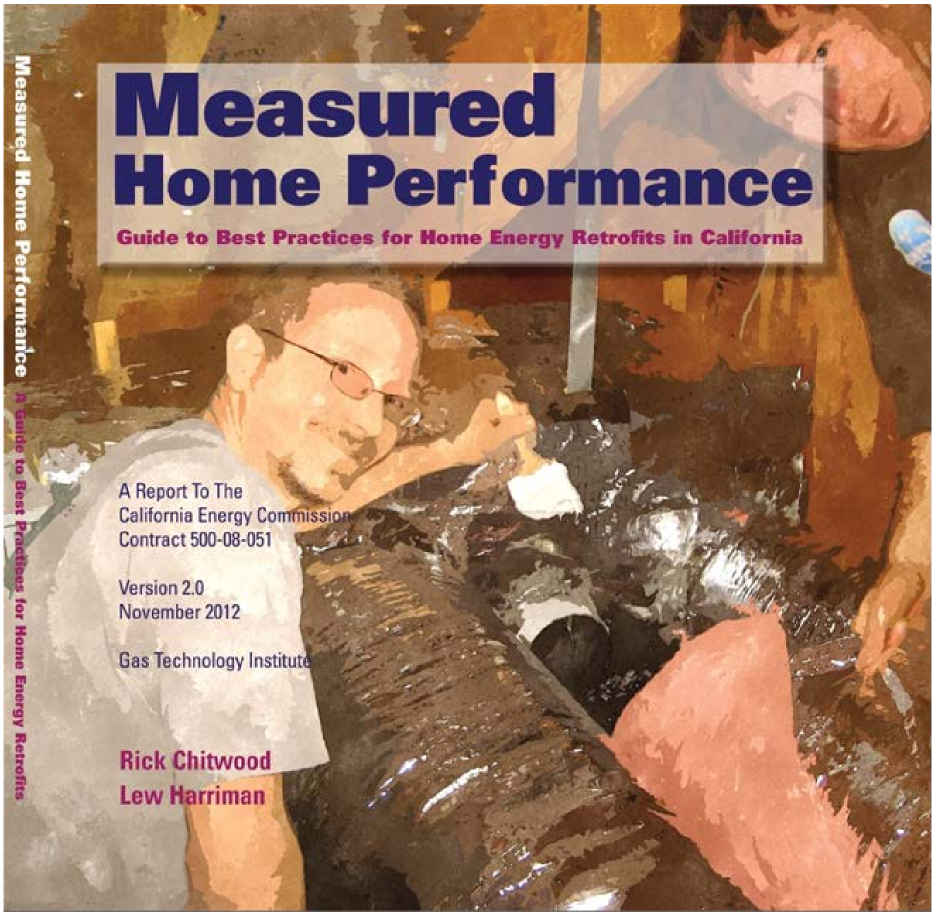


Photo Credit: Lew Harriman

Home Performance Contracting provides a homeowner with better comfort, better indoor air quality and a safer, more durable home which uses less energy. After an ideal home performance retrofit, the homeowner's monthly expenses are less—not more—than they were before the project, even after accounting for the added monthly cost of a loan which might be needed to fund it.

These are not small projects. To achieve monthly savings in heating and cooling costs large enough to make the project "self-funding," a home performance retrofit project will typically cost between $10,000 and $60,000. It usually replaces the entire heating and cooling system, including new ductwork. The new heating and cooling equipment will be less than half the size of the current equipment. The new air distribution system will be smaller, simpler and air-tight. The AC system refrigerant charge and the air flows will be measured and set.

The project will seal the complex construction assembly which separates the home from the attic, so that conditioned air cannot escape upwards into that attic. Then at the end of the project, the contractor will add insulation in the attic, burying the new sealed and insulated ducts so that heating or cooling capacity is no longer lost to the unconditioned attic.

Home performance projects also usually include replacing the home's water heater, pumps for well water or pool filtration and any lighting fixtures which penetrate a ceiling. Typical projects also upgrade the bathroom exhaust fans to near-silent units and provide a system which provides the home with filtered air for ventilation. Those are the usual hardware components of a project. If this sounds like a big, complicated project—it is. But the most important difference between individual component replacements and a home performance retrofit is that all of the critical energy features are redesigned and reinstalled together, as an integrated system.

The resulting home energy system is not only carefully engineered; it's also measured in its critical aspects as it is being installed—by the installers themselves. The installer's final "test-out" reports are an absolutely essential part of the project. These provide certainty that the building envelope and the HVAC components will work as a highly comfortable and energy-efficient system, in sharp contrast to the historically disappointing results of the traditional piecemeal approach to building houses.

Homeowners and contractors who are not familiar with home performance contracting are sometimes confused by the expanding number of different approaches to home energy conservation. This method is different in many respects, three of which are especially important to understand. With home performance contracting:

1. Installation quality is measured, not assumed. Measurements provide the feedback during installation which is so critical to finding and fixing the inevitable shortcomings. "Normal" shortcomings would double or triple energy consumption from equipment which should perform so much better than it typically does in the real world according to laboratory testing.
2. The work is done as an integrated whole—as one project rather than in pieces over several years. None of the components by themselves will achieve significant energy savings while also providing comfort, safety and indoor air quality. In fact, field measurements have shown that when such projects are "done in pieces" or when expected results are based on manufacturer's energy efficiency ratings alone, energy consumption can actually increase. Also, there may be increased risk from combustion appliances and reductions in indoor air quality.
3. The selection of the project components, and the integration of those components, is based on measured success from thousands of homes—not on hopeful estimates based on limited laboratory testing and modeling.

Measured Home Performance, Guide to Best Practices for Home Energy Retrofits in California, is a guide for home performance contractors, and an instructive reference for homeowners, who have an interest in deep energy retrofits for existing residential buildings in California. The techniques provided in the guide could also be applied to new construction at the design stage and to housing outside the state of California with the appropriate adjustments for climate variations.

The guide is structured in ten chapters to educate contractors and homeowners alike on potential savings, test-in and continuous testing requirements retrofit techniques, as well as tips and traps for a successful retrofit:

1. Introduction
2. Home Performance Contracting – What It Is, Is Not and Why
3. Typical Projects
4. Pre-visit Preparation
5. Visit 1 – Test-in
6. Tips and Traps for Proposals
7. Proposal Adjustment
8. Tips and Traps for HVAC
9. Tips and Traps for Air Sealing
10. Owner Education

The following pages are an excerpt from chapters eight and nine of the guide discussing good practice for HVAC and air sealing techniques.

Chapter 8
Tips and Traps for HVAC Design

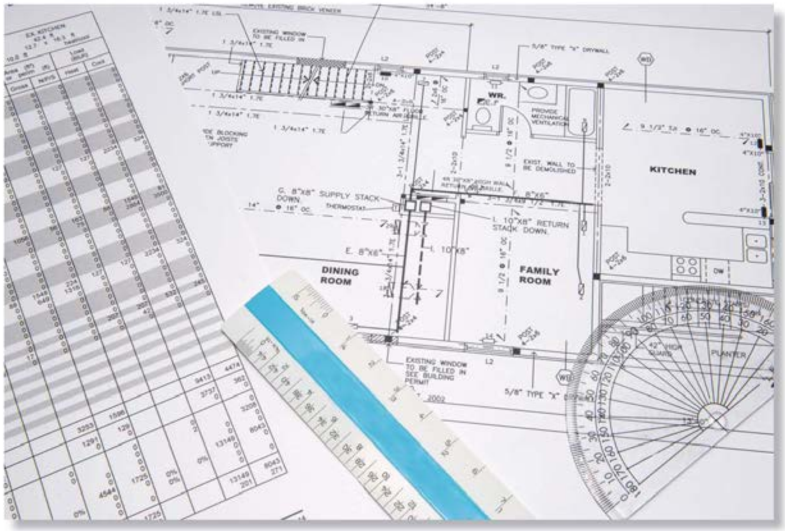


Fig. 8.1
Room-by-room load calculations are the foundation of excellent HVAC system design

Tips and Traps for HVAC Design & Renovation

A complete course in residential HVAC design is beyond the scope of this publication. HVAC design is a big topic. Critical details of implementation depend on the home, the budget, local fuel options and the type of system selected. In order to be helpful to the greatest number of readers, this chapter will focus on several best practices which are especially important for the most typical system used in California homes: the all-air, central heating and DX cooling system.

Better HVAC Design Goals and Assumptions

A perfectly-designed HVAC system has *just enough* heating and cooling capacity to keep up with the loads, provided that it's allowed to *run continuously*. In other words, it's a system with no excess capacity. It provides comfort without wasting either capacity or energy.

With a well-designed and properly installed HVAC system in a well-insulated and sealed thermal envelope, there's no need to set back the thermostat to save energy. In fact there's no need to touch the thermostat at all. The system just runs and provides comfort, no matter what the outdoor weather, year-round. Also, it's so quiet that the occupants may not even realize it's operating, especially since they never feel any uncomfortable blasts of either hot or cold air, because there's never any significant temperature difference between their ankles and their head. The house is comfortable all year long. So the homeowner simply never thinks about the HVAC system. It just works.

This is quite different from most of the HVAC systems installed in California homes. Traditionally, residential HVAC systems have been designed with too much cooling and heating capacity, because most designers have assumed (based on their bitter experience) that the building will be leaky and poorly insulated, and that the HVAC system will be installed badly. In other words, past HVAC design practice can be summarized as "When in doubt, use a bigger hammer."

In order for Performance Contracting to deliver reliable year-round comfort while also saving energy, it's critical to take a different approach; namely, design the HVAC system based *not* on failures and shortcomings of installation, but rather on what you *know* will be accomplished (and validated by in-process measurements) during the Performance Contracting project.

HVAC EQUIPMENT CAPACITY

Check Numbers for New Systems	Description
15,000 Btu/h per 1,000 ft² **(Conditioned space)**	**Target maximum installed heating capacity.** Lower Btu/h per 1,000 ft2 are better, because the higher Btu/h of furnace capacity needed to heat 1,000 ft2, the less effective is the current heating system. (Greater potential for improvement)
1 Ton per 1,000 ft² **(Conditioned space)**	**Target maximum installed cooling capacity.** Lower # of tons per 1,000 ft2 are better, because the higher the tons per 1,000 square feet, the less effective is the current cooling system. (Greater potential for improvement)
450 - 500 cfm per ton of AC system capacity	**Target minimum total supply air flow** Lower flow rates are not efficient in the dry California climate
50°F	**Lowest desirable coil-leaving air temperature** during cooling operation
100°F	**Highest desirable coil-leaving air temperature** during heating operation

Table 8.1 Check numbers for HVAC equipment capacity

Calculate Loads After Assuming Correct Installation of Energy Features

Among HVAC designers, there's a natural inclination to select equipment and design the system based on the assumption that duct work will leak, that the loads will be higher than expected and that the system won't be installed quite right. But with Performance Contracting, such major problems don't happen. So it's very important not to allow old assumptions about arbitrary load estimates and poor installation to sneak into the designer's thinking. If they do, the system won't work well, and it certainly won't save any energy.

Photo Credit: Lew Harriman

To avoid problems of poor comfort or excess energy use, check your load calculation assumptions and results against the values shown in table 8.1. More explanation of those numbers follows:

Total cooling load about 1 ton/1,000 ft²

If the total calculated cooling load is more than 1 ton per 1,000 ft² of occupied floor space in most of California, there's something wrong with either your assumptions or the scope of your project. Examine both closely. Adjust either your assumptions or your project scope, and then recalculate the loads. (The exception is in the desert, where you can expect higher cooling loads.)

Total heating load under 20,000 Btu/h/1,000 ft²

If the total calculated heating load is more than 20,000 Btu/h/1000 ft² of occupied floor space in most of California, there's something wrong with either your assumptions or the scope of your project. Examine both closely. Adjust either your assumptions or your project scope, and then recalculate the loads. (For most of the State, 15,000 Btu/h per 1,000 ft² of occupied space would be ample for a well-insulated air tight new house. The exceptions would be in snow country, where you can expect higher heating loads.)

Duct leakage near zero

The duct work and the duct connections will be air tight. The system won't lose more than 50 cfm total from the supply air flow through leakage at a test pressure of +25 Pa. In fact, a total leakage of 20 cfm_{25} and below is quite commonly achieved by well-trained crews.

Duct insulation with low losses

The duct work will be insulated to R-8, and then in the attic will be mostly buried by cellulose or glass fiber insulation. That level of insulation effectively reduces conductive duct losses to near zero for much of the year and only rises to the values shown in table 8.2 during the hottest and coldest hours.


Past practice - Poor insulation   R-40 Attic insulation - Measured & verified

Fig. 8.2 Design the HVAC based on insulation values you *know* will occur with Performance Contracting

Attic insulation that really performs

The attic will be insulated to R-40 and the crawl space (if there is one) will be insulated to R-19. The depth of the insulation will be verifiable quickly and easily, as shown in figure 8.2

Real-world walls and windows

Assume walls and windows will contribute to the loads according to their field-measured dimensions, locations and insulating characteristics—not according to an arbitrary assumption of single glazing at some guessed-at percentage of total wall area. The key characteristics of all the window assemblies will have been surveyed and recorded during the test-in visit.

Design the Air Distribution Based on Best Practices

In most new houses, air distribution design is an afterthought. The installers are expected to shoehorn the ducts into the home somehow, without creating too much chaos for other trades, and without any respect for, or understanding of, the critical tasks of commissioning and maintenance. But with Performance Contracting, air distribution

AIR DISTRIBUTION DESIGN

Check Numbers for New Systems	Description
500 - 700 fpm	**Minimum and maximum discharge velocity** from supply air diffusers
250 fpm	**Maximum velocity through return air grills and filters**
0.35" to 0.45" wc	**Duct design target - Max external static pressure for the air handler's fan.** (Maximum combined resistance of supply and return duct systems, including coils, filters, supply diffusers and return grills)
50 cfm_{25} or 5% of measured fan flow, whichever is less	**Maximum combined total air leakage from supply and return sides of the system.** (The real goal is zero air leakage, and 20 cfm_{25} is commonly achieved in practice by well-trained crews)
4,250 Btu/h per 1,000 ft² of occupied floor space	**Cooling load from attic duct work** Maximum conductive heat gain from attic duct work per 1,000 ft² of occupied space (Based on R-8 insulation, 40% of occupied floor space as duct surface, attic temperature of 140°F and supply air temperature of at least 55°F)
2,500 Btu/h per 1,000 ft² of occupied floor space	**Heating load from attic duct work** Maximum conductive heat loss from attic duct work per 1,000 ft² of occupied space (Based on R-8 insulation, 40% of occupied floor space as duct surface, attic temperature of 40°F and supply air temperature of no more than 90°F)

Table 8.2 Check numbers for air distribution design

design can take center stage, because the goals are lowest energy and greatest comfort (as opposed to highest speed and lowest bid).

These goals free the HVAC designer to use the best practices that have been known for decades and well-documented by ASHRAE, SMACNA and ACCA, but seldom implemented in the past. Some best practices for air distribution design are described below.

Duct runs can (and should be mostly) short and straight

This is possible because the supply ducts can go from the air handler to the nearest corner of each space—not way out to the windows.

SUPPLY AIR DUCT & GRILLE SIZING

Air flow (cfm)	Duct Diameter	Curved Blade Ceiling Grille[1]	Floor Grille[2]	High Sidewall Grille[3]
50	5"	4" x 4"	10" x 2.25"	4" x 4"
75	6"	6" x 4"	8" x 4"	6" x 4"
100	7"	6" x 6"	10" x 4"	8" x 4"
125	7"	6" x 6"	12" x 4"	10" x 4"
150	8"	8" x 6"	10" x 6"	8" x 6"
175	8"	10" x 6"	10" x 6"	10" x 6"
200	9"	8" x 8"	12" x 6"	10" x 6"
250	10"	14" x 6"	12" x 8"	10" x 8"
200	10"	10" x 8"	14" x 8"	12" x 8"

1. Typically a commercial-grade extruded aluminum grille
2. Simple bar-type grille (non-diffusing)
3. Typical residential double deflection type grille

Table 8.3 Check numbers for supply air ducts and grilles

Note that the grilles are the no-diffusing type. Leaving the grille, the air flow is straight or angled, but ***not*** *interrupted by the turbulance of either a diffuser or a damper. Dampers are at the supply plenum—not in the grille.*

With the improved enclosure, there's no need to send heating and cooling out to the windows. More important to comfort will be good air mixing in the space.

Supply air ducts & grilles sized using the "Goldilocks principle"

In other words, supply air ducts and grilles must not be too big or too small, but juuust... right. The check numbers in table 8.3 will help you make sure your supply duct and grille sizes are in the right ballpark. Within that range of grille sizes, you'll get the delivery velocity you need for good air mixing and comfort. That range of duct diameters is also pretty close to optimal. Any larger, and you'll have needless conductive losses from the extra duct surface area. Any smaller and the system will waste fan power pushing the air through the ducts.

Photo Credit: Lew Harriman

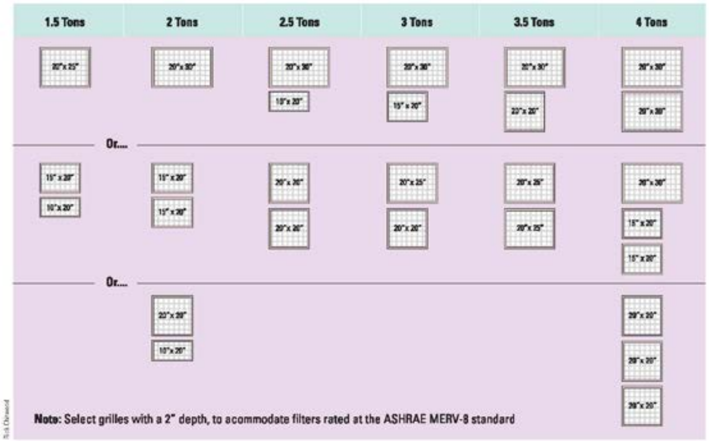


Table 8.4
Check numbers for return air grilles
These selections are based on standard, commercially-available hardware which meets two basic design criteria: less than 250 fpm velocity through the return air filter, which in turn keeps the presure drop through the grille and filter to less than 0.07" w.c.

Fast, straight air delivery - Supply air grilles without diffusers or dampers

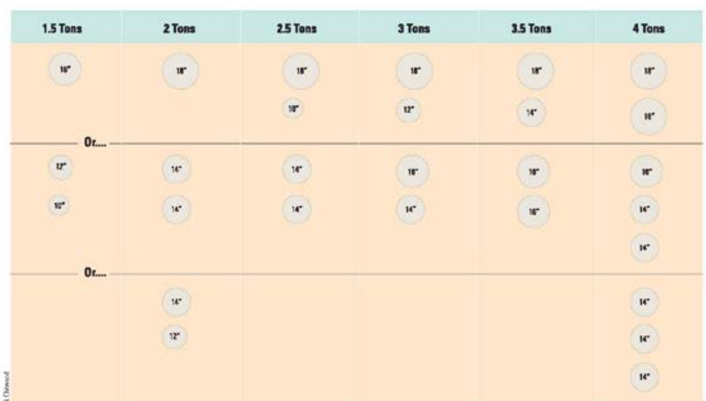
Good air mixing in the conditioned space is absolutely critical to comfort. You want even, nearly-equal temperatures top to bottom and side to side throughout the space. That's achieved when the fast-moving supply air stream pulls in room air at its edges, and blends that air with the warm or cold supply air quickly and blows it throughout the room (preferably across the room above head level).

If the grille has diffusers, the smooth, fast flow of supply air is destroyed, and the supply air sort of stumbles out ineffectively, cooling or heating only the area in its immediate vicinity. And if the grille has a damper, setting the flow with that damper can disturb both smooth flow as well as it's direction. You want the direction and speed to be constant and well-aimed, so that occupants don't get blasted with supply air.

Therefore, use supply grilles which keep the air moving straight and fast, without diffusers. And make sure that the necessary balancing dampers are installed *at the supply plenum*—not in the grille.

Placing the room's balancing damper deeper in the duct, away from the visually-obvious location of the grille, has the other advantage

Table 8.4
Check numbers for return air duct sizes



of discouraging the owner from fiddling with the flow after you have carefully balanced the system. (Uncomfortable owners sometimes try to get more air flow at the supply grille—they don't always realize that the usual problem is a dirty return air filter.)

Locate supply air grilles so they don't blow their high-velocity air onto room occupants.

Ideally we deliver supply air from the floor. Floor supply grilles provide excellent mixing in the heating season and very good room air temperature mixing in the cooling mode. However, many California houses have slab-on-grade foundations, which means floor grilles aren't an option.

When supply ducts are in the attic (the most common situation) there are two supply grille options. The best option is a high-sidewall supply grille. These deliver a high velocity plume across the ceiling. However, high-sidewall grilles require a vaulted ceiling with side wall access from the attic, or a dropped ceiling so that the duct can be brought down from the attic and set into a side wall. These architectural features are not typical in most houses.

So the most typical supply air grille is a true curved-blade extruded aluminum supply grille mounted in the ceiling. This grille delivers a smooth, high velocity air plume across the ceiling (not straight downwards) for reasonable mixing and no drafts on the occupants.

Photo Credit: Lew Harriman

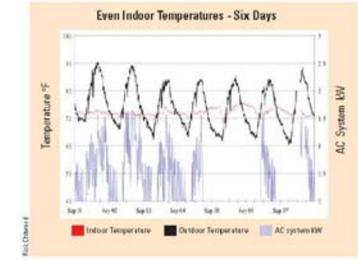


Fig. 8.3 Results from best practices
This house in Redding was designed and constructed using the best practices outline in this and other chapters. The homeowners didn't bother adjusting the thermostat... for six days. The system just works.

Don't design tiny ducts to small spaces

In small spaces like bathrooms, utility rooms and closets where load calculations call for less than about 75 cfm of supply air, don't bother running a separate supply duct. Just add the small space's load to the nearest large space, and upsize the airflow to the larger space accordingly. In a well-insulated and air tight home, there won't be any comfort impact. You'll save material, labor and fan power, while reducing conductive heat losses and gains from ducts.

For return grilles, bigger is better. But return ducts have to be "....juuust right"

The open area of return grilles and ducts must be large. Use the check numbers in tables 8.3 and 8.4 to ensure your design has enough open area at the return grill to allow smooth flow back to the air handler without excessive energy-consuming resistance at the grille.

Ducts, on the other hand, must not be *too* large, because they run through unconditioned spaces. When they are oversized, it's true that fan power is less. However, larger ducts also mean a larger surface area to lose and gain heat as the return ducts pass through unconditioned spaces. So for grilles, bigger is better. But for duct sizing follow the Goldilocks Principle and make them "...juuust right." ("Just right" sizing results in a combined total air flow resistance in the range of 0.35 to 0.45" for the supply and return duct systems, including all coils, filters, diffusers and grilles.)

The success of these best practices is quite remarkable. Indoor air temperatures are even and uniform, year-round. See figure 8.3 for one example, which shows a week of indoor vs. outdoor temperatures, along with the AC system's modest power consumption while achieving those results.

Design Calculations and Documentation Which Support Excellent Installation

The HVAC designer can help ensure excellent installation (and therefore excellent comfort) by the assumptions made during load calculations, and by the information on the load summary sheet.

Room-by-room load calculations

The heating and cooling loads must be calculated room-by-room, not by an arbitrary amount per ft^2. From room-by-room load calculations, the air flow to each space can also be easily calculated by the HVAC designer. With that information, the technicians can quickly measure and set the air flows needed for each space.

Constant year-round supply air flow rate

For trouble-free comfort, air flow and air mixing must be excellent all year round. Supply air grilles and diffusers need a constant volume to ensure good air mixing. *Therefore, the supply air flow rate must be the same for both heating and cooling seasons*. (The flow rate will probably be set by the cooling load, in most of the State other than in snow country.)

Moderate supply air temperatures year-round

Extreme supply temperatures waste energy and create extreme comfort problems. Calculate the required year-round air flow rate based on a supply air temperature of no more than 100°F for heating and no less than 50°F for cooling. These are moderate temperatures which save energy. Lower temperature differences between the duct and the surrounding spaces mean lower duct losses and therefore less energy use, as well as less stratification, better air mixing and therefore better comfort.

Fig. 8.4 Hydronic air handler, combined with a high-efficiency, sealed-combustion natural gas water heater

Equipment Selection Which Provides Comfort Without Energy Waste

Equipment selection is a bit more complex than just picking a furnace with a high combustion efficiency rating and a cooling system with an impressive SEER. But it's not really difficult, as long as the designer keeps a few key points in mind.

Combined hydronic air handler is an excellent choice

For several reasons, combined hydronic air handlers (also sometimes called "hot water furnaces") are an excellent choice for well-insulated, tight houses in most of California.

These units allow the use of a hot water heating coil for winter heat, which can be sized to match the low heating load without overheating the supply air. Also, the air handler can be equipped with a DX cooling coil which has a larger-than-usual surface area. The larger surface area reduces air flow resistance, which in turn allows larger-than-usual supply air flows and higher-than-usual supply air temperatures for cooling, which saves energy in the California climate for the reasons discussed in earlier paragraphs.

Finally, the "combined" part of the name comes from this equipment's ability to use hot water from the building's domestic hot water heater for HVAC heat. The hydronic air handler with its hot water heating coil and the domestic hot water heater are "combined" to form the HVAC system's heating equipment.

To understand why the combined hydronic air handler is such a good fit, it's important to understand the shortcomings of conventional furnaces and DX cooling equipment for the low loads that are typical of well-insulated, fairly air-tight homes in the relatively mild and dry California climate. There are problems in California for conventional furnaces even though that equipment is so economical and works so well in many other parts of the country:

a. Furnaces simply have far too much capacity. They will "short-cycle," switching on and off frequently, delivering too much and too little heat, rather than a smooth flow

Photo Credit: Lew Harriman

of *just enough* heat to keep up with the loads. In most of the State, you won't have heating loads of more than about 15,000 Btu/h per 1,000 ft² of occupied space. The smallest available conventional central system furnace has about 40,000 Btu/h of capacity... *more than twice* the correct size for a well-built 1500 ft² home.

b. Conventional furnaces don't work properly unless they have a relatively high outlet temperature. If the supply temperature leaving the furnace is less than about 110°F, it means the temperature rise through the furnace is too low. That in turn means that too much heat remains in the furnace. So it shuts down, because you don't want to burn up your furnace. The problem is that for best comfort, and energy efficiency we *don't want a supply air temperature higher than about 100°F.* Conventional furnaces can't provide that low a supply air temperature without damage to their heat exchangers.

c. If supply air temperatures are higher than about 100°F, air in the room will stratify severely. Hot air rises. Very hot supply air rises quickly to the ceiling and stays there. It does not mix well with cooler air closer to the floor. So while your head is hot, your ankles are cold. Therefore, for better comfort and less energy waste we need moderate supply air temperatures during the heating season. These really can't be provided by conventional furnaces.

Combined hydronic air handlers are also a good fit for the customization of the cooling system which is so important in the California climate. Without the humidity loads that are typical in the rest of the country, in California we can maintain comfort using more air flow per ton and higher supply air temperatures for cooling than would be typical in other States.

d. In a humid climate, an air flow rate of 350 or even as low as 300 cfm/ton is a good choice, because the low flow allows deeper cooling and more dehumidification. But in California, we don't need or want that much dehumidification. It would waste energy and create uncomfortably dry conditions, because the system will be running for *long* periods. Instead, we need larger air flows and less-deep cooling. A supply air flow rate near 500 cfm/ton is a better selection for energy and comfort in our climate. And a supply air temperature of 60°F is better than 50 or 55°F.

e. But here's the potential problem. With such a high flow rate across the coil, what we save in cooling could be chewed up by resistance to air flow—unless the coil has a larger face area than would be typical in other parts of the country. See the check values in table 8.5 for ideal sizes of cooling coils compared to the total load on the system.

f. This important customization explains why the combined hydronic air handler is also a good choice on the cooling side of the system. The air handler allows better customization—larger cooling coils—rather than the smaller standard "A" coils. These have a smaller air flow capacity per ton, since they are designed to match the air flows through conventional furnaces.

Air-source heat pumps are another good choice

One of the traditional limitations of air-source heat pumps has been that they don't produce *really* high temperature supply air during winter operation. But for well-insulated and reasonably air tight homes, that's not a bug... it's a feature. We want 100°F or less.

Heat pumps make sense where there is no natural gas available for heating. Natural gas is generally the most economical fuel for hot water and for HVAC heat. But it's not available everywhere in California, and the alternative of propane-fired heat is very costly. So the all-electric, high-efficiency heat pump can be a useful alternative to the combined hydronic air handler in areas without natural gas service.

COOLING COIL SIZE	
System Size (Tons)	Evaporator Coil Size (Tons)
1.5	2 to 2.5
2	2.5 to 3
2.5	3.5 to 4
3	4 to 5
3.5	5
4	5

Table 8.5
Check numbers for A/C coil sizes

Another potential alternative would be ground-source heat pumps, which have received a great deal of attention nationwide. But these really don't have advantages in the mild California climate. Compared to air-source units, ground-source heat pumps are quite expensive to buy, install and maintain. And unlike other parts of the country, without long hours at extreme temperatures and humidities, in California there are no significant compensating advantages to the limitations of ground source heat pumps in single-family homes.

500 cfm/ton... OK, but how? Use a larger evaporator coil

As discussed earlier, the target range for air flow through the AC system is about 500 cfm/ton. That's difficult to achieve if one uses the manufacturer's usual recommendations for matching the evaporator coil to the condensing unit.

The more energy-efficient choice is to use "the next size up" evaporator coil for the load. The condensing unit and its compressor stay the same... but the evaporator coil is larger than usual. The larger evaporator coil allows air flows near 500 cfm per ton without excessive pressure drop.

The values in table 8.5 provides useful check numbers for pairing the right size evaporator with the load on the condenser.

Limit super-high-efficiency filters to special situations

Excess filter pressure drop kills overall system efficiency. That resistance to air flow costs a lot in motor horsepower, especially since the well-designed HVAC system has very long run cycles.

It's helpful to educate the customer on this subject. Most consumers know they should have the oil and oil filter changed every so often in their car. They would also benefit by knowing that *monthly air filter changes* is an excellent way to ensure that comfort and energy savings continue after the system has been started up.

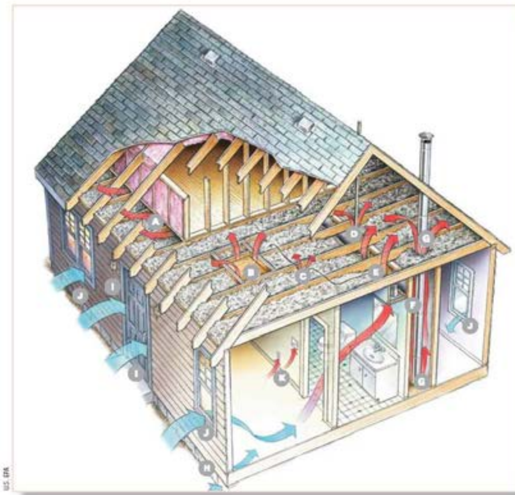
Typical modern HVAC systems are designed to accommodate very high-grade, MERV-8 pleated filters. Those filters remove more than 70% of particles between 3 and 10 microns in diameter—a huge reduction. These can be installed in the deep filter racks built into the return air grilles recommended in this chapter.

But if the client's home is located next to a highway or is in a dusty agricultural region, or if it has a particulate-sensitive occupant, the home may benefit from super-high-efficiency particulate removal at the MERV-11 level. *In those cases, be sure to select a supply fan speed which provides full air flow at a pressure high enough to overcome the resistance of that high-efficiency filter.*

For the more typical home (without sensitive occupants and without a nearby constant source of large amounts of fine particulate) the owner will save energy by avoiding super high-efficiency filters.

Photo Credit: Lew Harriman

Chapter 9
Tips and Traps for Air Sealing The Enclosure



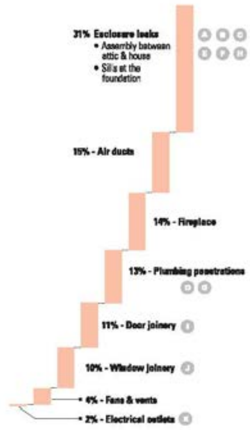


Fig. 9.1 Leak-hunters guide
The drawing show many of the typical locations of air leaks. The graph shows the probable percentage of annual air leakage in each location, based on computer modeling of a typical 1600 ft², single-story house assumed to be in Modesto.

Sealing the Enclosure

Sealing the enclosure is a very important part of the Performance Contracting project. It requires careful planning and in-process air tightness measurements. Careful planning is really critical, because if you're not careful a tight house can create serious life safety and moisture problems. And in-process air tightness measurements are also critical because if progress isn't being made, you might be sealing the wrong holes and wasting your effort. Here's a logical sequence for the air sealing part of your Performance Contracting project:

1. Don't proceed until you have:
   a. Removed any vermiculite insulation and/or rodent and pest infestation or droppings from the attic.
   b. Cleared and cleaned the attic (and crawl space, if there is one) for safe access to all potential leak locations.
   c. Reworked the exhaust ducts and properly terminated any fans or combustion appliances which may have originally exhausted to the attic.
   d. Fixed any roof leaks (or water accumulation in the crawl space, if there is one).
   e. Tested all natural draft combustion appliances for safety under normal and worst-case depressurization conditions and made any necessary repairs or replacements to ensure safe operation in an air-tight home.
2. Establish the target sealing rate.
3. Duplicate the test-in air tightness test to establish the baseline air tightness value.
4. Locate the leaks.
5. Seal big leaks before small leaks, and high and low leaks before middle-level leaks.
8. Run the blower door test about once an hour to measure progress towards the air tightness target.

Fig. 9.2 Fix moisture problems before air sealing begins
There's no point in trying to air-seal the house until after the major problems are fixed.

Pre-sealing Preparation & Safety Issues

Air-sealing the home should not be approached casually. If the home has moisture or indoor air quality problems, sealing the building can make these problems much worse. And in the case of combustion appliances, some problems can even become life-threatening.

Safely remove vermiculite insulation

Vermiculite is a naturally occurring mineral used in construction and gardening products. It looks like shiny, small pieces of popcorn, and is usually light-brown or gold in color. Vermiculite is still mined and distributed for a number of uses, including insulation.

Photo Credit: Lew Harriman

This lightweight, granular mineral insulation is rare in California. But unless it's new it must be removed. Here's why. 70% of vermiculite produced before 1990 came from a single mine in Libby, Montana. That vermiculite was contaminated with a small amount of asbestos. Much of the Libby vermiculite was used as attic insulation. It was sold under the product name "Zonolite." The Environmental Protection Agency estimated in 1985 that 940,000 American homes contained Zonolite attic insulation.

Vermiculite mined today for use in insulation is from sources considered to be free of asbestos contamination. However, unless the vermiculite is known to be new and free of asbestos, it must be removed before you can safely work in the attic. Undisturbed older vermiculite is not believed to be a serious problem. But during your project, you will definitely be disturbing it, moving it and spreading dust. Older vermiculite dust from the Libby, MT mine may contain asbestos, which is a breathing hazard for both workers and occupants. If you need more detailed information about vermiculite, consult the U.S. EPA website: http://www.epa.gov/asbestos/pubs/verm.html

Clear and clean the workspace

Attics and crawl spaces are favorite homes for rodents and insects. So any infestation or droppings must be removed before you start creating airborne dust with your air-sealing activities.

Also, it's important to clear out any stored materials or belongings that get in the way of safe access to all the corners and edges of the spaces you'll be working in. You don't want to strain muscles or put a foot through a ceiling because you're trying to avoid disturbing clutter in an already confined and awkward space.

Make sure exhaust ducts and vent stacks don't vent inside the attic

Venting into the attic is never, ever appropriate or allowed by codes. But if "informal" workers or homeowners have inadvertently terminated exhaust ducts or vent stacks inside the attic, these must be extended out and terminated properly outdoors, before you begin your project.

If you wait until later, you'll just have to come back and seal up any new penetrations and re-do your final blower door test. So it's best to fix these problems before you begin air sealing, to ensure safe working conditions and to avoid costly rework.

Fix water leaks and moisture accumulation problems

See figure 9.2 Water intrusion problems like damp crawl spaces and roof leaks can sometimes go on for years without generating mold or odor problems. But after the home is tightened, the free flow of drying air that keeps moisture from accumulating will be gone. When the home leaks water, moisture problems can become more severe quite quickly, after the home is tightened up. So it's important to make sure that the roof does not leak, and that any leaks around windows or doors are fixed.

The same is true in the crawl space, if there is one. Make sure it's well-drained and has a vapor barrier installed over the ground before you tighten up the house. Also your crew will thank you for creating a more reasonable work environment in the crawl space. (Or they won't thank you—because it's still not much fun to work down there—but at least you'll know you've done the right thing!)

Test natural draft combustion appliances and fix them if necessary

Finally, and most important: *If there are any problems with the combustion appliances, you must fix them or replace them, before tightening the house.* You can create a life-safety risk if you have a tight house with an appliance which is generating carbon monoxide.

Your test-in safety testing will alert you to any potential problems. That's one of the main purposes of the extensive test-in visit. So if there are combustion appliance problems, fix them before you tighten the home.

Establishing the Ending Sealing Rate Target

Establish your *ending sealing rate* target before you begin, so that you'll know when it's time to stop. In any project, you'll eventually reach a point where the sealing progress slows down so much that it is no longer cost-effective to try to find more leaks to seal.

Typical sealing rate targets are between 100 and 400 cfm_{50} per crew-hour. In other words, if after an hour of work a well-trained and well-motivated crew has not been able to reduce the air leakage rate by (a value that you've established between) 100 to 400 cfm_{50}, then it's probably time to stop sealing and work on other parts of the overall project.

Ideally of course, we'd like to seal every home completely airtight. But in the real world you'll need to stop after you've sealed everything which is accessible, or after the crew's progress falls below your sealing target rate. Otherwise the air sealing phase would be so expensive that nobody could afford it. Also keep in mind that if you *are* able to seal below a certain point (less than 0.35 ach_{50}) you will need to either recommend or add ventilation to the HVAC part of your project.

Duplicating the Test-in Air Tightness

The first step in the actual air sealing project itself is to set up the blower door exactly as it was done during the test-in visit, and then run the test until you have duplicated the leakage rate measured during that test-in visit. That will be your baseline.

Leave the blower door test setup in place as you work, so that about each hour or so, you can re-test the air leakage, to measure your progress towards the target sealing rate.

Fig. 9.3 Blower door testing is repeated periodically during air sealing operations to provide feedback to the crew about their progress towards their air tightness target.

Photo Credit: Lew Harriman

Locating & Fixing Typical Attic Air Leaks

Locating the leaks is relatively easy at the beginning, and then becomes progressively more difficult. You can begin by looking through the attic for the big holes and missing pieces of ceilings that are typically found there. Keep in mind the nearly obvious guidance that says: "find and fix big leaks before small leaks." Figure 9.1 showed some of the typical leak locations. Here is some additional discussion of each location.

Kitchen and bathroom cabinet soffits

In most houses, the tops of cabinets end at reachable hand-height, rather than going all the way up to the full ceiling height. Between the top of the cabinet and the ceiling level is often a "soffit." This is gypsum wall board formed into a wall between the top of the cabinet and the ceiling. But often, that soffit is open to the attic at the top, and sometimes also open to the room at it's base, just above the top of the cabinet.

That air gap can be large, as shown in figure 9.4a. To fix it, you'll need to block it off with gypsum board or some form of insulation board which can support the weight of the additional insulation you'll be installing later. Then the edges of that board will need to be sealed with caulking, or with minimal-expansion foam.

Fig. 9.4a
Open soffit, above bathroom closets

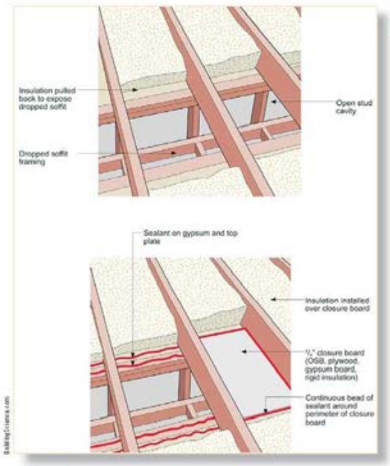


Fig. 9.4b
Air sealing an open soffit

Fig. 9.5
Open stud bays which penetrate the attic

Fig. 9.6
Attic access hatches

Fig. 9.7
Sealing HVAC duct penetrations in the attic

Open stud bays

Sometimes, as shown in figure 9.5 above, the tops of interior stud bays project into the attic and are not capped. These can be stuffed with a folded batt of fiberglass insulation to act as backing, and then covered with a layer of spray foam to ensure a tight air seal.

Attic access hatches

Attic access hatches are seldom insulated (and sometimes missing). Every attic access gets three improvements:

1. Insulation, to at least R-19 with three (3) inches of foil-faced foam board (Polyisocyanurate).
2. Weather stripping, to limit air leakage, and..
3. A 10" to 15" dam on the attic side, to keep attic insulation from falling into the home when the access hatch is opened.

Duct chases and shafts

Anywhere the supply and return ducts of the HVAC system penetrate the attic, the opening must be sealed up. In the case of small gaps, polyurethane foam-from-a-can is probably the preferred method. If there are larger gaps, then add a collar of gypsum board or sheet metal to narrow the gap before applying foam.

Photo Credit: Lew Harriman

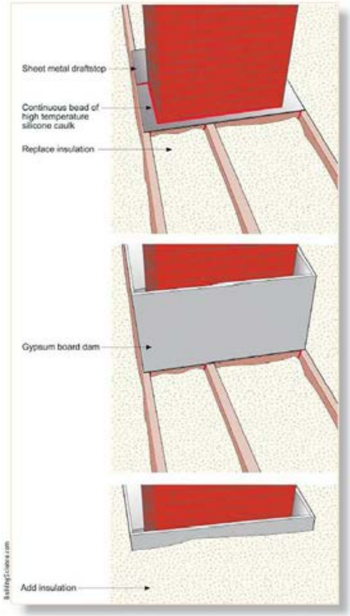


Fig. 9.8
Sealing a chimney penetration in the attic

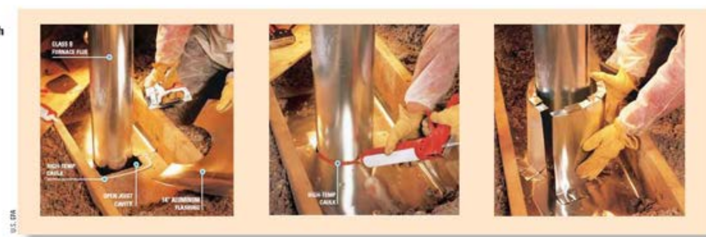
Chimney penetrations

A chimney will nearly always have a wide gap around it, to make sure that combustible material is not in contact with the hot flue. The wide gap is the easiest way to meet that code requirement.

When you seal this gap, keep in mind the need to prevent combustible material from contacting the flue. First fit a sheet metal collar to close the gap. The use a fire-rated sealant to caulk the seam between the sheet metal collar and the masonry, as shown in figure 9.xx. Be sure the joints are well-covered with sealant, because your next step will make those joints more difficult to fix later, if they leak air.

After the gap is air-sealed with the collar and caulking, build a vertical insulation dam out of gypsum board, taking care to keep a 2" air space around the masonry, as shown in figure 9.8 at left. The air space protects the insulation—which may be combustible—from coming in contact with the hot masonry.

Fig. 9.9
Sealing a furnace flue which penetrates the attic



Combustion exhaust stacks

Combustion exhaust stacks are like chimneys—except that they are made of metal and therefore conduct heat better than masonry. So it's important to keep the insulation away from their hot surface.

This is done in two steps, just like chimneys. First, fit a collar to the lower part of the stack, where it enters the attic. Seal the collar against air leakage, using fire-rated sealant or caulking. As with the chimney collar, be sure the joints are well-covered with sealant, because the next step will make those joints more difficult to fix later, if they leak air.

Next, make and install a metal insulation protection collar as shown in figure 9.9 above. Seal the bottom of the metal collar to the collar which seals the stack penetration, so that insulation can't accidentally leak under the metal collar into the air gap between the collar and the hot stack.

Fig. 9.10 Sealing plumbing and electrical penetrations

Plumbing & electrical penetrations

Plumbing penetrations are nearly a given in attics. Seal them with foam or caulk. There is no need to be concerned about stack temperature with plumbing or electrical boxes, so standard minimal-expansion foam is quite adequate for the task.

Photo Credit: Lew Harriman

Can lights

Can lights which penetrate the ceiling need to be sealed up, air tight. Usually, replacing the fixture with a modern air-tight fixture rated for full insulation contact is the quickest and most economical way to ensure both air tightness and safety. But sometimes, circumstances dictate that you make combustion-safe covers for the can lights, and seal the covers to the ceiling with spray foam.

In those cases, keep in mind that older can lights were designed for air flowing upwards through the fixture to keep the wiring safely cool. So encapsulating such fixtures with an insulated, combustion-safe box is an uncertain solution. It may not provide adequate safety against overheated wiring if the owner keeps using high-wattage bulbs in the fixture. In those cases, equipping the fixture with a compact fluorescent bulb is at least a nod towards a safer fixture. But replacement is definitely the preferred option.

Fig. 9.11a
Insulated, air-tight cover for a can light which penetrates the attic

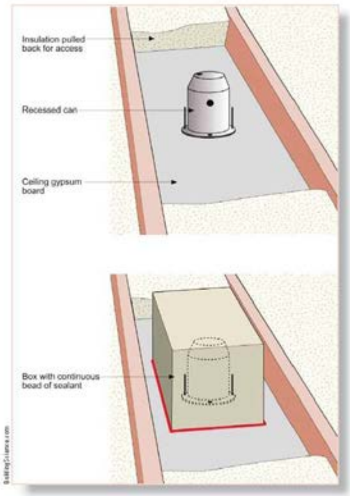
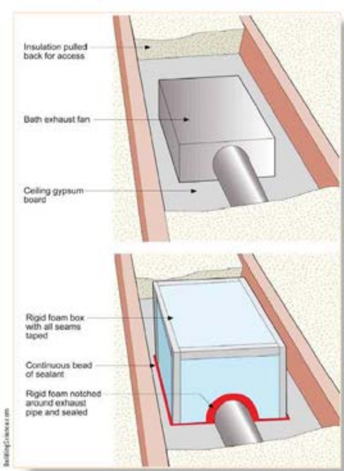


Fig. 9.11b
Diagram of an insulated, air-tight cover for a can light

Bathroom exhaust fans

Exhaust fans set into the ceiling of the upper floor often have sheet metal enclosures which leak a great deal of air into the attic. If these are not replaced with modern, more air-tight and more silent units, be sure to seal up the seams of the fan enclosure in addition to the gap around the fixture where it is set into the ceiling.

Fig. 9.12
Diagram of an insulated, air-tight cover for a leaky bathroom exhaust fan



Ceiling gypsum board seams and wall top plates

The gypsum board which forms the walls and ceiling of the floor below the attic has seams which leak air at both the base of the wall, and at the top of the wall where it enters the attic. Seal up all of the top plates of the interior walls, plus the attic-side seams of the ceiling gypsum board, as seen in figure 9.13 below.

Fig. 9.13
Sealing the top plates of interior walls and the seams of ceiling gypsum board panels with spray foam
Also note the foam-covered, air-sealed electrical box

Photo Credit: Lew Harriman

Fig. 9.14 Many plumbing penetrations from the crawl space
Also note the open joints of the tongue-and-groove subflooring

Fig. 9.15 Lots of penetrations plus tongue-and-grouve subflooring?
The best plan may be to simply install spray foam over the entire surface

Air Leaks From The Crawl Space

Typical leaks from the crawl space to the 1st floor of the house are very much like the penetrations and air leaks in the attic. Seal up the joints around plumbing, electrical and duct penetrations.

Also in the crawl space, there are often openings where the floor joists connect to the exterior walls. To seal those big openings, there are two classic techniques. For many identical large openings, you can cut pieces of board insulation to fit, and then seal up all four edges with minimal-expansion foam. For smaller or more irregular openings, or opening which have pipes or wires running through them, you can stuff the opening with folded glass fiber batt insulation. Then cover the insulation with a layer of spray foam to ensure a tight seal.

Another classic leak problem in older California homes is tongue-and-groove subflooring, installed at a diagonal to the exterior walls. The joints between the boards open over time and leak a great deal of air upwards, into vertical wall cavities and other floor-to-ceiling paths like pipe chases, duct chases and chimney penetrations. Usually, there are so many penetrations and odd structural members supporting the floor that the most practical way to seal so many seams is simply to spray foam over the entire crawl-space side of the floor. That approach also allows for an excellent air seal around all plumbing, HVAC and electrical penetrations.

Progress Testing with the Blower Door

As you seal the home, for the first hour or two the major leaks will still be visually obvious to a trained crew. But after the obvious leaks are closed up, use the blower door every hour or so, to see how much work remains. The last blower door test, which shows that the target sealing rate has been reached, should be recorded as the test-out value. (Note: This will be an impressive improvement, and therefore worth mentioning and explaining to the owner.)

Fig. 9.16
Thermal cameras can help locate less-obvious air leakage points
Be careful, however, to choose a camera with an adequate thermal sensitivity (NEDT of 100 milliKevin or less).

Finding Less-obvious Gaps, Cracks and Holes

In every home, there are gaps, cracks and holes which are simply very tough to locate, even for the most experienced of crews. That's why thermal cameras are such a useful tool when air sealing houses.

As shown in figure 9.16, a thermal camera can help you locate the areas where air is getting into the building. It shows surface temperature patterns. And often, those patterns indicate locations where cold or warm air from outdoors is leaking into the building.

Of course to be effective, a thermal camera needs two things: a significant temperature difference between indoors and outdoors, and a negative pressure high enough to pull outdoor air into the building. Without those, it's unlikely that the thermal camera will show you enough of a pattern to help you locate the problem area.

Use the blower door to create the usual negative pressure inside the home, so that outdoor air will be pulled in through any leaks. Let the blower door run for about 10 to 15 minutes, so that the infiltrating air will have enough time to generate a visible temperature difference on the walls or ceiling near the leak location.

And if the outdoor temperature is nearly the same as the indoor temperature, you may have to run the AC system or the heating system indoors for an hour or so to create a visually useful temperature difference. The classic rule of thumb is that there should be a temperature difference of 18°F between indoors and outdoors for best results. But that old standard really misses the point. The useful temperature difference is simply one which generates a useful visual pattern. And that can be achieved at temperature differences as small as 5°F between indoors and outdoors, depending on your camera. The better your camera, the less temperature difference it will need to show you a useful pattern.

A word about thermal cameras for Performance Contracting

The camera needs to be sensitive enough, and must have enough pixels to show the thermal pattern of leakage. These days there is a wide range of thermal cameras available at lower and lower prices. But not all of the lowest-price cameras are useful for finding air leaks in buildings, even though they may be well-suited to inspections of electrical panels, where temperature differences are much greater.

If you're considering using a thermal camera to help with air leak detection, make sure the camera has a thermal sensitivity of less than 100 milliKelvin degrees. And ideally, the thermal sensitivity should really be 70 milliKelvin or below.

In specification sheets, this critical figure of merit is variously described as the "noise equivalent delta T (NEDT)", or "thermal sensitivity." And the value is usually displayed as either thousandths of a degrees C or whole degrees milliKelvin. (Degrees Kelvin are the same as degree Celsius, except that the Kelvin scale begins at absolute zero while the Celsius scale goes positive above the freezing temperature of water.)

The minimal useful thermal sensitivity or NEDT you'll need for detecting air leakage patterns under less-than-ideal temperature differences between indoors and outdoors will be shown on data sheets as either 0.01°C or 100 degrees milliKelvin. Values above this level are a warning that the camera won't be useful under many of the small temperature differences between indoors and outdoors that you're likely to encounter in daily operations.

Photo Credit: Lew Harriman

5.2.2 Measured Home Performance Upgrades in Field Test Sites

For the two field test sites, Chitwood Energy met with the homeowners, conducted a pre-upgrade audit, scheduled the work to follow the installation of the radiant systems, and then performed the work. A radiant HVAC system including a high-efficiency water heater and a chilled water storage cooling system was installed in both houses prior to the application of MHP techniques.

Installation on the Grandstaff house (Figure 29) was conducted on October 19-20, 2011 and the 6th Avenue house on October 21-23, 2011. For the Grandstaff house, air sealing (including weatherstripping), insulating and sealing ducts, adding attic insulation, and adding a ventilation fan in accordance with ASHRAE 62.2 requirements were the primary measures applied.

For the 6th Avenue house (Figure 31), knob and tube wiring was found in the attic space, so foam insulation was cut and inserted below the wiring. In this house, the existing floor furnace was isolated from the crawl space, air sealing (including weatherstripping), and crawl space wall and floor measures were applied. A ventilation fan was also installed in this house.

The installers notes, provided below, show why these two research houses are excellent examples of the need for and barriers to measured home performance.

5.2.3 6th Avenue Field Test House

5.2.3.1 Original Envelope Conditions (built in about 1930, approximately 1,000 square feet, raised floor):

1. About 70 percent of the house is wired with 80 year old knob and tube wiring.
2. Balloon framing (wall cavities are open to the attic space).
3. Multiple wall cavities open to the attic and crawlspace.
4. High rates of air infiltration, 2,840 CFM_{50}, 20 ACH_{50}. This high rate of air infiltration was measured after the following areas had been sealed:
    - The fireplace damper was closed,
    - A 5" x 14" opening next to one of the window air conditioners was sealed with tape, and
    - The combustion air opening to the attic above the water heater was sealed.
5. The original air infiltration rate was much higher than the measured infiltration rate. The sealing goal is about 1,200 CFM_{50}.
6. No ceiling insulation.
7. No wall insulation.
8. No floor insulation.
9. Single hung vinyl windows with high solar low-e coating.

5.2.3.2 Improvements performed on October 21st, 22nd, and 23rd:

1. Confirmed the initial air infiltration measurement (2,840 CFM_{50}).
2. Replaced the dirt clogged bug screen on the front and rear gable end vents with ¼″ mesh hardware cloth.
3. Removed all debris from the attic.
4. Installed a Panasonic Whisper Green bathroom exhaust fan (80 CFM) in the drop ceiling area above the tub/shower. Bathroom ventilation will help control humidity and reduce condensation on the cooling panel.
5. Sealed the wall cavities that are open to the attic with foil faced foam board and moisture cure gun foam. On the front half of the house, more of the upper ceiling boards were removed to access the lower ceiling.
6. Constructed an insulation dam (2″ x 8″ lumber) at the attic access hatch located in the utility room ceiling. Weather striped the access hatch. Insulated the hatch with foil faced R-14 foam board.
7. The radiant heating/cooling piping in the attic was re-routed low enough to be covered by the loose fill ceiling insulation (no more than 8″ above the lath and plaster).
8. Installed 1 ½″ foil faced foam board for ceiling insulation. The foam board was cut into strips and placed between the ceiling joists. The foam board insulation did not cover the knob and tube wiring, eliminating the need to replace the wiring now. The foam board has an R-value of about R-10 (but is not continuous since it will be installed between the ceiling joists).
9. Replaced the dirty and torn bug screen on the existing crawlspace vents with ¼″ mesh hardware cloth.
10. Air-sealed the floor assembly from the crawlspace. Special attention will be required at the bathtub and the air leaks into the exterior, balloon framed, wall cavities. Re-measure the air infiltration rate after sealing the floor.
11. Removed the debris, especially organic debris, from the crawlspace and installed a six mil polyethylene sheet vapor retarder on the soil, to reduce moisture levels in the house.
12. Sealed the crawlspace vents and other air leaks between the crawlspace and outside will provide a more thermally buffered space below the floor. Installed a crawlspace vent fan that will exhaust 40 cfm continuously (0.02 cfm/ft^2).

Figure 86: Clearing Debris from 6th Avenue

Photo Credit: Gas Technology Institute

Figure 87: 6th Avenue Hydronic Tubing

Photo Credit: Gas Technology Institute

Figure 88: Foil Faced Foam Insulation below Wiring

Photo Credit: Gas Technology Institute

5.2.3.3 Additional improvements to consider:

1. Insulate the exterior walls of the home using high density loose fill cellulose insulation (fill-tube method of installation). Note: Before the walls can be insulated the knob and tube wiring needs to be replaced. The insulation will be installed from the outside and the penetrations to the stucco will be patched and painted to match.
2. Repair the toilet. The toilet is running constantly which will keep the toilet tank cold and could cause condensation puddles on the floor.
3. Check the gas range for carbon monoxide production. Consider a carbon monoxide detector or a range hood.
4. Check refrigerant charge and adjust the TXV to assure at least 5°F of subcooling. Install a mixer in the storage tank to provide enough velocity over the evaporator to prevent ice build-up. Other options might be greater tube spacing on the evaporator heat exchanger, and/or installing a smaller condensing unit.
5. Provide written operating instructions and an occupant walk-through.

5.2.3.4 Final testing:

The final air infiltration rate was 1,578 CFM_{50} with window air conditioners sealed.

5.2.4 Grandstaff Drive Test House

5.2.4.1 Existing Envelope Conditions (built in about 1972, approximately 1,000 sq. ft., slab-on-grade):

1. Reasonable air infiltration rate, 1,520 CFM_{50}, 11 ACH_{50}. Air infiltration was measured as the house was found. Measured air leakage was higher than it should have been since the radiant panel installation was in progress and the ceiling penetrations for the supply and return tubes to each panel had not been sealed yet. The sealing goal is about 1,000 CFM_{50}. The existing furnace, which is located in a vented hall closet, will remain.
2. R-15 ceiling insulation, about four inches of loose fill cellulose.
3. Viewing the walls in infrared indicated that wall insulation was present.
4. Vinyl windows with low solar low-e coating.

5.2.4.2 Improvements performed on October 19th and 20th:

1. Confirmed initial air infiltration measurement (1,520 CFM_{50}).
2. Constructed an insulation dam (2″ x 8″ lumber) and install new ¼″ mesh hardware cloth at the hall furnace closet combustion air inlet through the ceiling.
3. Improved the latch and air sealed (weather stripping) on the hall furnace closet door.
4. Constructed an insulation dam (2″ x 8″ lumber) around the discharge of the whole house fan.
5. Installed a gravity closure over the whole house fan discharge using 1 ½″ foil faced foam board.
6. Constructed an insulation dam (2″ x 8″ lumber) at the new attic access hatch located in the hallway ceiling. Weather striped the access hatch. Insulated the hatch with foil faced foam board – R-14.
7. Installed foil faced foam board eve vent baffles. Installed ¼″ mesh eave screens at the eave vents.
8. Sealed the old attic access hatch in the closet ceiling. Insulate over the hatch with R-38 loose fill insulation.
9. Insulated the existing duct system with foil faced (FSK) fiberglass duct wrap R-6.
10. Sealed the ceiling assembly air leakage with moisture cure gun foam. Areas that were focused on are top plate leaks, radiant heating and cooling tube penetrations, and the area around the kitchen range exhaust duct.
11. Air sealed other locations in the home that showed infiltration during the infrared scan; the top door trim on the front door, the top door trim on the door to the garage, and the weather stripping on the door to the garage.
12. Insulated the ceiling to R-38 with loose fill insulation.
13. Replaced the existing bathroom exhaust fan with a Panasonic Whisper Green bathroom exhaust fan (80 CFM), duct out with 6″duct. The new bathroom exhaust fan will provide some whole house ventilation and spot bathroom ventilation will help control humidity and reduce condensation on the cooling panel.

5.2.4.1 Final Testing:

The final air infiltration rate was 803 CFM_{50}

5.2.5 Results and Recommendations

These two proof of concept demonstration projects provided examples of the difficulty of MHP implementation.

Insulation Strategy Impacted

At the 6th Avenue house, a house built in the 1930's, knob and tube wiring was found. It is poor practice to bury knob and tube wiring in thermal insulation. There is a slight chance that burying the wiring in insulation might cause overheating, and the thermal insulation always increases the difficulty of replacing the wiring in the future. To employ MHP the insulation contractor should have an electrician experienced in knob and tube replacement as part of the team. Since a suitable electrician could not be located a more expensive insulation and air sealing strategy was used – installing 1.5" foil-faced foam board in the attic (R-10) below the knob and tube wiring. This method makes it possible to replace the wiring in the future and yet provides good attic insulation and an air tight pressure boundary.

Oversizing

The radiant heating/cooling system was installed in the 6th Avenue house before any thermal insulation was installed or any air sealing was performed. Without any insulation in the floor, walls, or ceiling the newly installed cooling system met the loads for several hot weeks before the ceiling insulation was scheduled for installation. With a completely insulated envelope the loads will be reduced by a factor of 4. Proper MHP implementation would create better coordination between the mechanical installation and the envelope improvements.

Mechanical System Installer Insulating Piping Penetrations

The piping manifolds for the radiant heating/cooling system at Grandstaff were all located in the attic – and were extensive since the manufactured panels used were small. As the mechanical installer connected pipes he also spent time packing loose fill attic insulation into each piping-penetration hole. To provide a more effective air barrier and to assure that hot humid attic air would not penetrate the ceiling assembly the insulation installer removed the loose fill insulation and installed closed cell foam in the piping-penetrations. MHP, implemented in the proper sequence with the radiant system installation, would have eliminated this duplication of work.

5.2.5.1 Results of Air Sealing

The 6th Avenue house air infiltration was reduced to 1578 CFM_{50} from 2840 CFM_{50} using the blower door, and the Grandstaff house air infiltration was reduced to 803 CFM_{50} from 1520 CFM_{50}. Leakage was reduced by 44 percent and 47 percent, respectively. The energy reduction is expected to bring the peak energy demand by the houses within the capacity of the radiant heating and cooling system once complete (see note on 6th Avenue house).

5.2.6 Key Takeaways

Key takeaways from this work on the integration of Radiant HVAC systems and measured home performance are:

1. Radiant heating and cooling systems are fully compatible with measured home performance techniques.
2. Hydronic tubing should be installed as close to the bottom chord of the ceiling trusses as is possible so they can be covered with insulation.
3. Holes where the system penetrates the ceiling, sidewalls, or other air barriers should be sealed carefully with foam sealant.
4. Knob and tube wiring should be replaced prior to application of the radiant hydronic system.

Figure 89: Grandstaff with Insulated Attic

Photo Credit: Gas Technology Institute

CHAPTER 6: Technology Transfer

6.1 Introduction

This project component focuses on connection with the market to disseminates best practices and other findings from the research to HVAC professionals, installers, and consumers in California. The objective of the project is to develop classroom training, multimedia online materials, and consumer-grade information materials based on results of the radiant cooling, heating, and measured home performance tasks. A critical element of this project is developing all of the materials in an easily understandable fashion to the identified audience, i.e., reduce the technical, engineering data to a format that builders, architects, and others can understand without losing the core of the information.

6.2 Objective

The objective of this project is to disseminate best practices and other findings from this program to HVAC professionals, installers, and consumers in California through several venues:

1. Develop classroom, online, and consumer-grade materials based on project results
2. Conduct classroom training sessions at participating utility sites
3. Host a multimedia website containing subject expert videos and other materials
4. Provide consumer-grade materials to participating utilities in a suitable format

6.3 Results

The results of the project included the following deliverables:

1. A softcover book *Measured Home Performance, Guide to Best Practices for Home Energy Retrofits in California* by Rick Chitwood and Lew Harriman, is available on Amazon.com.
2. Syllabus and four training classes with three California investor-owned utilities
3. A consumer's guide to measured home performance
4. A contractor's guide to measured home performance
5. 18 short videos featuring measured home performance techniques
6. A website: http://www.measuredhomeperformance.com containing the videos, a link to the softcover book for free download, and the consumer's guide and contractors guides
7. Technical papers in several forums

6.3.1 Classroom Training

Training is designed for inclusion in the participating utilities' energy efficiency program offerings. The use of utility training, rebates, public awareness campaigns, and system demonstrations combine to accelerate market acceptance. For this project, training was sponsored by and conducted at three utility partners: PG&E, Sempra, and SCE.

Classroom training materials were developed by Doug Beaman and Associates and Chitwood Energy. The MHP guide was supplied to the students as a textbook and a syllabus in the form of a 3-ring binder was also provided. The one-day class used the following schedule:

Table 35: Training Class Schedule

Measured Home Performance - Best Practices Guide			
Time		Topic	Description
9:00	10:30	Opportunities for Improvement	Review current building and HVAC practices and show the inefficiencies in typical systems: including duct leakage, duct design, infiltration, and HVAC equipment sizing
10:30	11:15	Case Studies	Review case studies showing the current state of building practices and the potential for high performance homes.
11:15	12:00	Radiant Cooling	Review of the two radiant cooling systems installed in Sacramento homes during 2011.
12:00	1:00	Lunch	
1:00	3:00	Performance Factors	Review the most important performance factors for improving home energy efficiency including: Reducing infiltration, doors & windows, insulation installation quality, HVAC systems.
3:00	3:45	*Measured Home Performance: A Guide to Best Practices for Home Energy Retrofits in California*	Review of the book with explanations of how to use it and how to get additional copies.
3:45	4:30	How Do We Get There?	How can the principles reviewed today be impletemented both in new construction and in the retrofit industry in California.

Source: Gas Technology Institute

The syllabus was organized into the following sections:

Tab 00 Table of Contents

Tab 0 Introduction

Tab 1 Opportunities for Improvement

Tab 2 Case Studies

Tab 3 Radiant Heating and Cooling Installations

Tab 4 Performance Factors

Tab 5 Measured Home Performance – Book Overview

Tab 6 The Road Forward

Classroom training was very well attended and well received by the students. There were 149 total attendees and both the Stockton and Irwindale classes were filled to capacity. Table 36 provides the location of the four classes, the sponsoring utility, the number of students and information on the quiz score.

Table 36: Classroom Training Classes Conducted

Training Date	Location	Sponsoring Utility	Number of Students	Average Quiz Score (if applicable)
2/24/12	Stockton, CA	PG&E	32	n/a
3/13/12	Downey, CA	Sempra	20	half had 100% on quiz
7/26/12	Irwindale, CA	SCE	88	87%
7/27/12	Tulare, CA	SCE	9	80%

Source: Gas Technology Institute

6.3.2 Consumer and Contractor-Grade Materials

The focus of the consumer brochure was to develop and produce a consumer-grade information booklet to assist homeowners in understanding:

- What a radiant HVAC system is, how it works, and what is different about it
- How large the energy and carbon emission benefits of radiant systems can be
- What integrated energy design and installation is
- How it works
- Why it is beneficial and to what extent those benefits can be measured and guaranteed
- How it compares to other alternatives for home energy savings investment;
- How the consumer can locate and select a capable and reliable contractor

The focus of the contractor brochure was to:

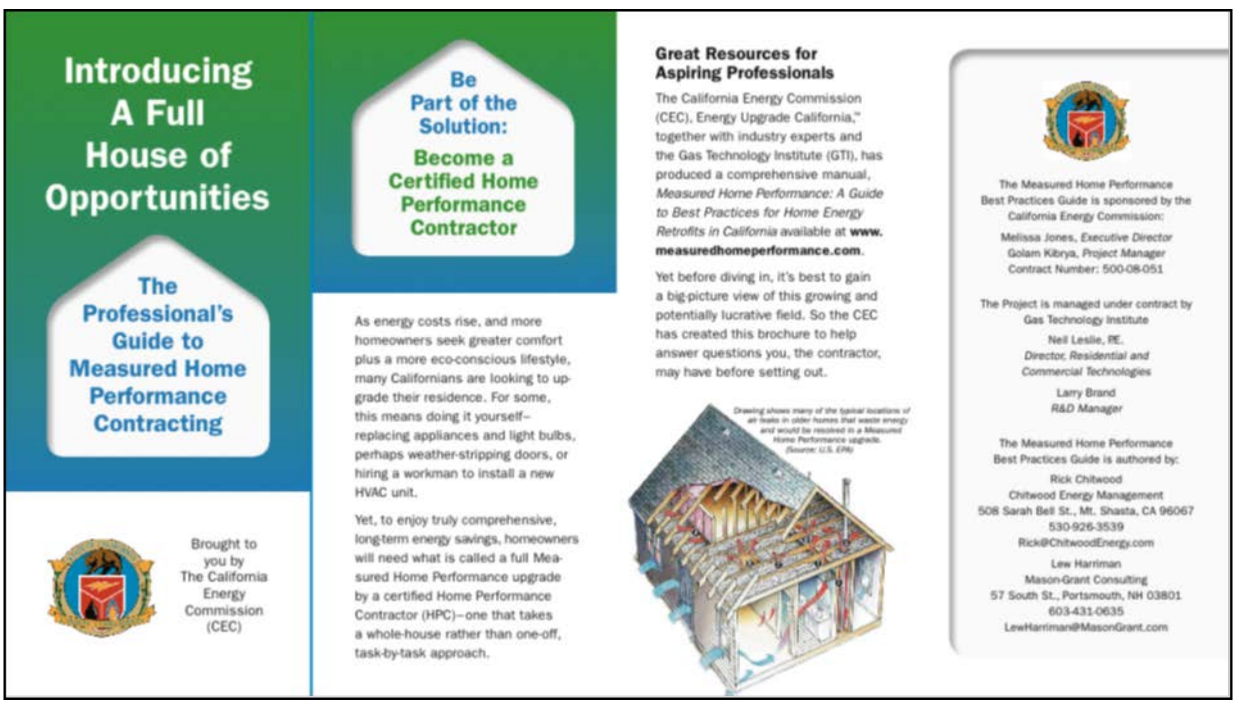
- Provide a definition of Measured Home Performance Contracting
- Explain why it matters (e.g. right thing for many projects, homeowners looking for total solutions and learning about incentives available for those solutions, potential income for contractors);
- Include information for professionals

Consumer-grade and contractor grade materials were developed by J-U Carter and are available at the website: http://measuredhomeperformance.com/. Files suitable for printing and distribution by utility partners are available from the principals. Figure 91 and Figure 91 show snapshots of those materials.

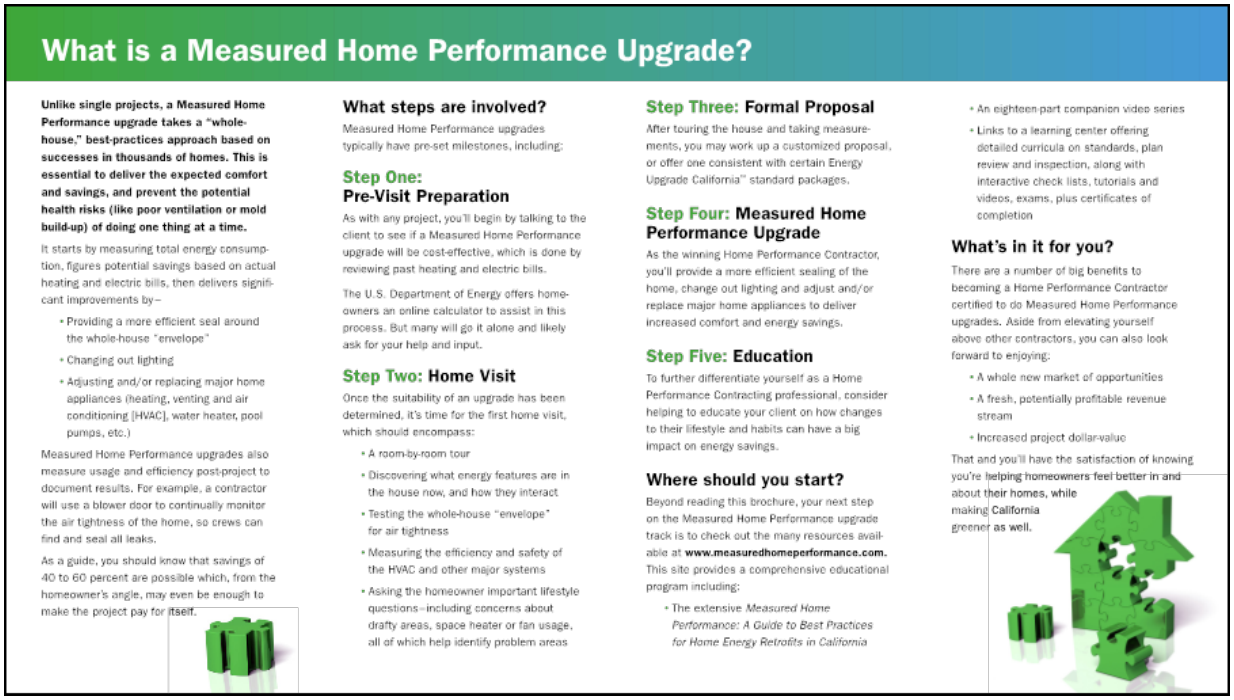
Figure 90: Consumer Brochure

Source: J-U Carter

Figure 91: Contractor Brochure



Source: J-U Carter



What is a Measured Home Performance Upgrade?

Unlike single projects, a Measured Home Performance upgrade takes a "whole-house," best-practices approach based on successes in thousands of homes. This is essential to deliver the expected comfort and savings, and prevent the potential health risks (like poor ventilation or mold build-up) of doing one thing at a time.

It starts by measuring total energy consumption, figures potential savings based on actual heating and electric bills, then delivers significant improvements by–

- Providing a more efficient seal around the whole-house "envelope"
- Changing out lighting
- Adjusting and/or replacing major home appliances (heating, venting and air conditioning [HVAC], water heater, pool pumps, etc.)

Measured Home Performance upgrades also measure usage and efficiency post-project to document results. For example, a contractor will use a blower door to continually monitor the air tightness of the home, so crews can find and seal all leaks.

As a guide, you should know that savings of 40 to 60 percent are possible which, from the homeowner's angle, may even be enough to make the project pay for itself.

What steps are involved?

Measured Home Performance upgrades typically have pre-set milestones, including:

Step One: Pre-Visit Preparation

As with any project, you'll begin by talking to the client to see if a Measured Home Performance upgrade will be cost-effective, which is done by reviewing past heating and electric bills.

The U.S. Department of Energy offers homeowners an online calculator to assist in this process. But many will go it alone and likely ask for your help and input.

Step Two: Home Visit

Once the suitability of an upgrade has been determined, it's time for the first home visit, which should encompass:

- A room-by-room tour
- Discovering what energy features are in the house now, and how they interact
- Testing the whole-house "envelope" for air tightness
- Measuring the efficiency and safety of the HVAC and other major systems
- Asking the homeowner important lifestyle questions–including concerns about drafty areas, space heater or fan usage, all of which help identify problem areas

Step Three: Formal Proposal

After touring the house and taking measurements, you may work up a customized proposal, or offer one consistent with certain Energy Upgrade California™ standard packages.

Step Four: Measured Home Performance Upgrade

As the winning Home Performance Contractor, you'll provide a more efficient sealing of the home, change out lighting and adjust and/or replace major home appliances to deliver increased comfort and energy savings.

Step Five: Education

To further differentiate yourself as a Home Performance Contracting professional, consider helping to educate your client on how changes to their lifestyle and habits can have a big impact on energy savings.

Where should you start?

Beyond reading this brochure, your next step on the Measured Home Performance upgrade track is to check out the many resources available at **www.measuredhomeperformance.com.** This site provides a comprehensive educational program including:

- The extensive *Measured Home Performance: A Guide to Best Practices for Home Energy Retrofits in California*
- An eighteen-part companion video series
- Links to a learning center offering detailed curricula on standards, plan review and inspection, along with interactive check lists, tutorials and videos, exams, plus certificates of completion

What's in it for you?

There are a number of big benefits to becoming a Home Performance Contractor certified to do Measured Home Performance upgrades. Aside from elevating yourself above other contractors, you can also look forward to enjoying:

- A whole new market of opportunities
- A fresh, potentially profitable revenue stream
- Increased project dollar-value

That and you'll have the satisfaction of knowing you're helping homeowners feel better in and about their homes, while making California greener as well.

Source: J-U Carter

6.3.3 Other Outreach Activities

The availability of the guide was communicated to many organizations including links to the multimedia website in their literature or emails. A complete list of organizations contacted is as follows:

- Air Conditioning Contractors of America
- Air-Conditioning, Heating, and Refrigeration Institute
- American Council for an Energy Efficient Economy
- American Society of Heating, Refrigeration and Air-Conditioning Engineers (ASHRAE)
- American Subcontractors Association (ASA)
- Associated Builders and Contractors (ABC)
- Associated General Contractors of America
- Associated Specialty Contractors
- Bay Area Chapter, ASA, Inc.
- BIA (Building Industry Association) of Central California
- BIA of the Bay Area
- BIA of the Delta
- BIA San Diego
- Builders Exchange of Alameda County
- Builders Exchange of Stockton
- Builders Exchanges

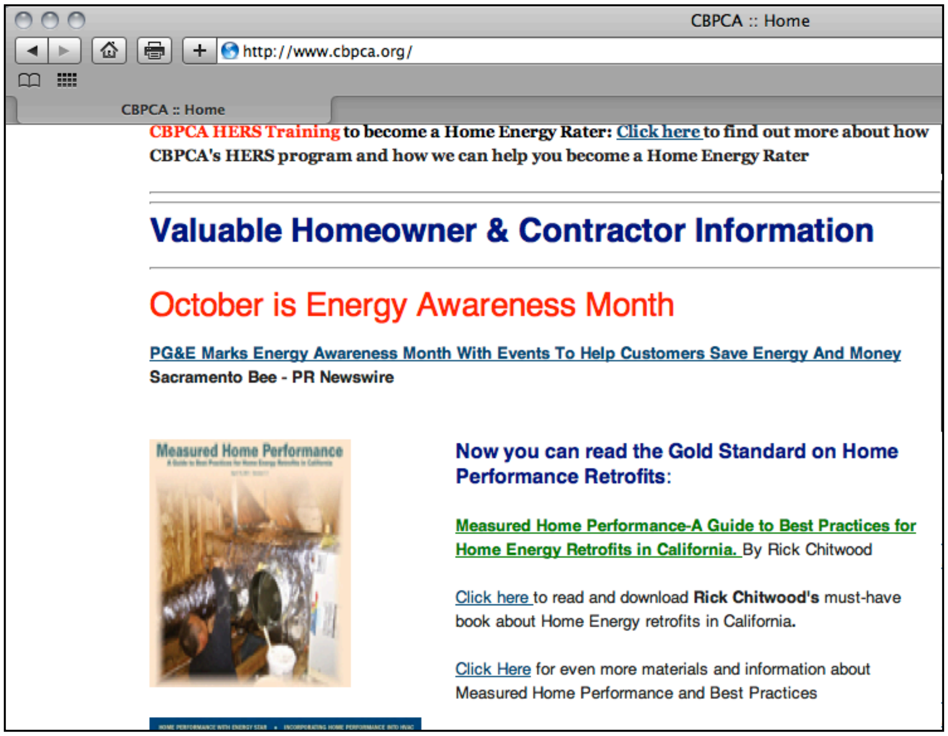
- Builders' Exchange of Santa Clara County
- Builders' Exchange of the Central Coast Inc.
- BuildingGreen, Inc.
- California Building Industry Association (CBIA)
- California Building Performance Contractors Association
- California Professional Association of Specialty Contractors
- California Sheet Metal and Air Conditioning Contractors National Association (Cal SMACNA)
- Capital City Chapter, ASA
- Central CA Chapter of ABC
- Construction Industry Roundtable
- Constructor Magazine
- Contra Costa Builders Exchange
- Energy Upgrade California
- ENR California
- Golden Gate Chapter of ABC
- Golden Gate Chapter, ASHRAE
- Golden State Builders Exchanges
- High Performance Buildings Magazine
- Home Builders Association of the Central Coast
- Home Builders Association of Tulare and Kings Counties
- Humboldt Builders' Exchange, Inc.
- Kern County Builders Exchange
- Kern County Home Builders Association
- LA/Ventura Chapter of ABC
- Marin Builders Association
- Mechanical Contractors Association of America
- National Association of Women in Construction
- National Contract Management Association
- North State BIA
- Orange Empire Chapter, ASHRAE
- Peninsula Builders Exchange
- Placer County Contractors Association
- Plumbing-Heating-Cooling Contractors Association of California (PHCC)

- Redwood Empire Chapter of ASA
- Redwood Empire Section, ASHRAE
- Refrigeration Service Engineers Society - CARSES (California Association of the Refrigeration Services Engineer Society) Regional Association
- Sacramento Regional Builders Exchange
- Salinas Valley Builder Exchange (Central Coast Builders Exchange)
- San Diego Chapter, ASHRAE
- San Francisco Builders Exchange
- San Joaquin Chapter, ASHRAE
- San Jose Chapter, ASHRAE
- San Luis Obispo County Builders Exchange
- Santa Barbara Contractors Association
- Santa Maria Valley Contractors Association
- Shasta Builders Exchange
- Sierra Delta Chapter, ASHRAE
- Solano-Napa Builders Exchange
- Southern California Builders Association
- Southern California Chapter, ASHRAE
- The North Coast Builders Exchange
- Tri-County Chapter, ASHRAE
- Tulare and Kings Counties Builders Exchange
- Valley Builders Exchange
- Valley Contractors Exchange
- Ventura Contractors Exchange
- Women Contractors Association

Specific action was taken by the following organizations.

- The California Building Performance Contractors Association posted a link to the MHP guide on its website (Figure 93). The organization also sent the PDF of the MHP guide to the 70 students who had attended training in Pasadena and included the link in eNewsletter to several thousand members.

Figure 92: CBPCA Website with Link to Best Practices Guide



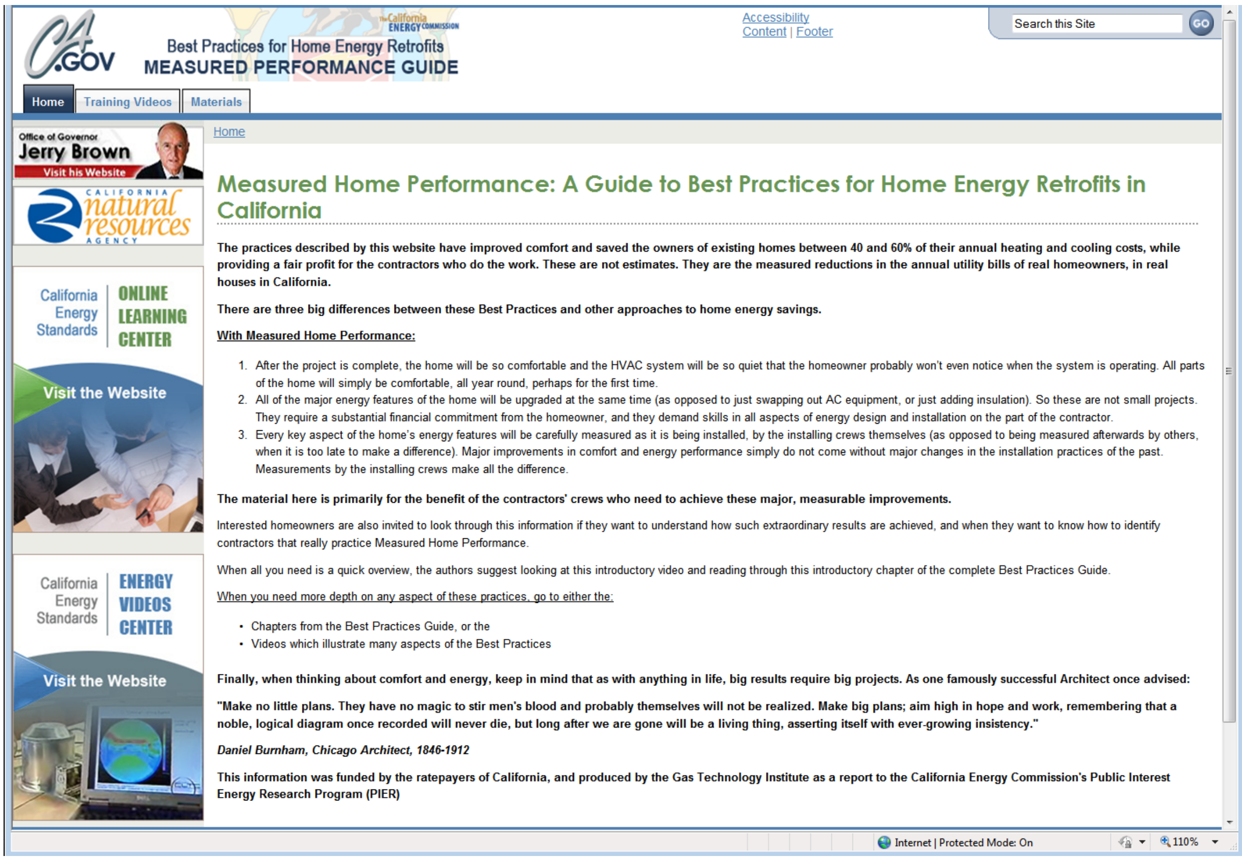
Source: Gas Technology Institute

- The Building Industry of the Bay Area included the link in an email to approximately 1,700 members in March 2012.
- The Refrigeration Service Engineers Society embedded a link to the MHP guide in an article J-U developed for them. This was distributed in their April 2012 eNewsletter.
- The Bay Area Chapter of ASA included a link in their eNewsletter to approximately 400.
- The Builders Exchange of San Luis Obispo County included the link in an online bulletin to approximately 675 members. BE Humboldt included it in an email to approximately 300 members and mentioned in a weekly newsletter. North Coast Builders Exchange posted a pdf of the MHP guide on their website. Santa Clara confirmed distribution in newsletter to approximately 700. Santa Clara most recently requested, and received, a modification of the article for possible future use.
- The Associated Builders and Contractors, Inc. Northern California requested and received the press release, contractors' brochure and links and planned to distribute in their eNewsletter.
- The California Building Industry Association (CIBA) requested the contractor's brochure for review but has not confirmed links or any potential use.

6.3.4 Multimedia Website

A multimedia website at http://measuredhomeperformance.com/ was developed in this project, as shown in Figure 93. The website is based on Energy Commission design and has links to Energy Commission resources. The website contains PDF versions of the guide, the consumer brochure, and contractor brochure. The 18 training videos are also hosted on the website. Google analytics allows one to be able to see who has viewed the video, if they have viewed the video to completion, how many repeat users are viewing the videos, and how many times they have watched a particular video.

Figure 93: Multimedia Website



Source: California Energy Commission

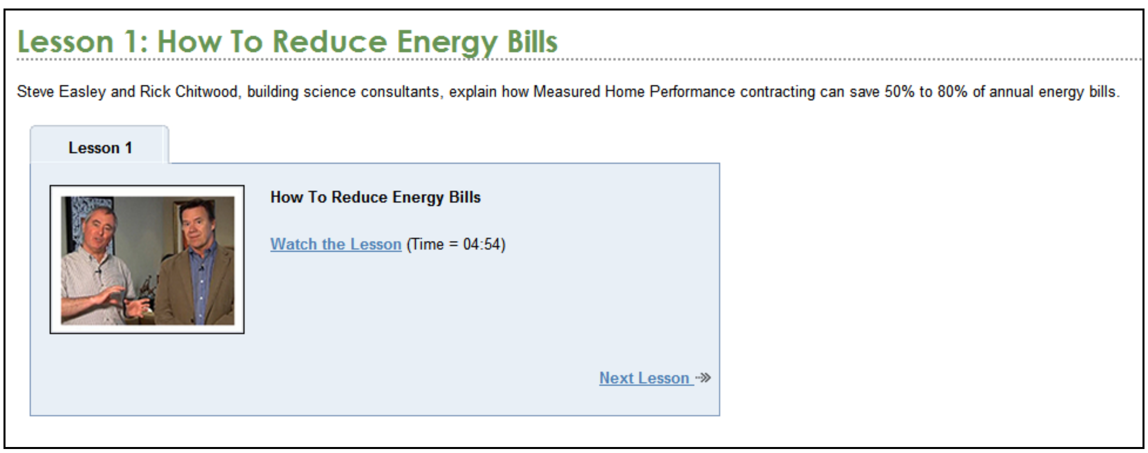
Online training materials were developed by the team and include 18 three to five minute videos covering the MHP guide. The topics covered in the videos include diagnostic tools, infrared thermography, and other basic concepts. The videos were filmed in a casual setting. They serve as a supplemental piece for people to refer to for the basic training. Titles for the 18 videos are as follows:

1. 1: How To Reduce Energy Bills
2. 2: House as a System
3. 3: Dispelling Energy Myths

4. 4: Utility Bill Disaggregation
5. 5: Orientation and Site Inspection
6. 6a: Blower Door Testing
7. 6b: Insulation Assessment with Infrared Camera
8. 7: Air Sealing and Insulating Attics
9. 8: Air Sealing and Insulating Crawl Spaces
10. 9: Air Sealing and Insulating Walls
11. 10: Heating System Equipment
12. 11: Assessing Air Flow in HVAC Systems
13. 12: Assessing Air Conditioning Performance
14. 13a: Assessing HVAC Duct and Distributions Systems
15. 13b: Evaluating Airflow and Room Air Delivery
16. 14: Electric Base Load
17. 15: Consumer Education
18. 16: Tips for Home Performance Contractors

A screenshot of the first lesson is provided in Figure 94

Figure 94: Training Video Lesson 1



Source: Gas Technology Institute

6.3.5 Technical Papers

Several members of the team developed technical papers and presentations on the topic of measured home performance to improve awareness within the technical community. A list of the papers and presentations is provided below:

Chitwood, Rick and Lew Harriman, ***Measured Home Performance Best Practices for Home Energy Retrofits***, ASHRAE Journal, January 2012

Harriman, Lew, *Does Any of This Actually Work?* presented to the National Energy Efficiency Technology Roadmapping Summit, Portland, Oregon, September 2012

Harriman, Lew, *Measured Home Performance; Moving Residential Energy Consumption Closer to Net Zero*, 2013 ASHRAE annual meeting seminar paper, January, 2013

Chitwood, Rick Mike McFarland, and Lew Harriman, *In-Process Measurements Are Better than Beefy Codes and 3rd Party Verifiers – Measured Home Performance in California*, ACI National Home Performance Conference, May 2013

CHAPTER 7: Summary and Conclusions

Results from the cooling season field test show that the systems performed extremely well. The capacity of the storage tanks was never exhausted during peak hours, allowing the almost complete elimination, other than the water circulation pump, of peak energy use which when quantified was reduced by 95 percent. The shifting of compressor use to night-time allowed further savings, calculated at 19 percent, due to a higher COP at lower ambient temperatures compared to a non-storage system. Humidity issues were not found – spot ventilation the bathrooms and kitchen appears to have adequately removed internally generated moisture and the system was never shut down by the dew point control. Temperature stability was better with the radiant system than the forced air system, and this was achieved without sacrificing a reasonable pull down rate for initial cooling, which was measured at an initial 3°F per hour.

Results from the heating season field test showed that the radiant heating system performed very well. The A. O. Smith Vertex 76,000 Btu/hr input capacity water heater with 96 percent thermal efficiency was sufficient to meet the load of the Grandstaff and the 6th Avenue houses during the winter. The radiant panels performed well in both houses. The design conditions of 15 Btu/hr/ft^2 of radiant heat from the panels at 120°F and 0.3 gallons per minute of hot water per panel were confirmed in these studies based on the circuiting used in the design. Heating season energy savings at Grandstaff was 34 percent compared to its baseline and in the 6th Avenue house, savings was 57 percent when compared to its baseline. The average heating season savings was 45 percent for the two houses in the Sacramento area with improved comfort. A predicted savings for the increase in efficiency of the water heater alone would yield a 15 percent savings for Grandstaff house and 30 percent savings for 6th Avenue house, leaving approximately 25 percent savings to be spread between thermal envelope improvements, the performance of the radiant heating system and the use of a lower thermostat setpoint at each location.

Economic analysis of the cost of traditional HVAC systems and the radiant system design show an incremental cost of approximately $3000 at full market when installed in new construction and factoring in the savings from the elimination of ductwork. Energy savings for cooling alone of approximately 20 percent, supported by the field test results, yields a payback of between five years and 15 years without peak-shifting incentives from the utility in California climate zones 10 and 12. With utility incentives at $1000/kW for peak load shifting, this payback period drops to two to five years. Adding incremental heating system efficiency improvements can reduce energy costs between 10 – 45 percent, depending on the starting point, and improve the payback for the system and using integrated installation techniques with measured home performance significantly reduces the energy consumption of the house since less chilled water storage is required.

Training in measured home performance and radiant heating and cooling systems was conducted at PG&E at their Stockton learning center, by Sempra in Downey, and by SCE in Irwindale and Tulare. Approximately 150 attendees learned about the latest techniques to provide quality installation while simultaneously measuring the effectiveness of the upgrade, and received a briefing on the project. Feedback from attendees was overwhelmingly positive and dissemination of measured home performance information was successful.

Finally, technology transfer for this project took several forms to reach a variety of audiences:

1. A softcover book *Measured Home Performance, Guide to Best Practices for Home Energy Retrofits in California* by Rick Chitwood and Lew Harriman is available on Amazon.com.
2. A consumer and contractor's brochure, 18 short videos, and four training classes
3. A website: www. Measuredhomeperformance.com containing the videos, a link to the softcover book for free download, and the consumer's guide and contractor's brochures
4. Technical papers in several forums

The authors conclude that radiant heating and cooling systems have an excellent potential for energy savings if the following conditions are met:

1. Utility incentives for peak-shifting are critical to cover the cost of the system above traditional HVAC equipment.
2. Measured home performance must be implemented along with the radiant technology in order to minimize the load and the size of the chilled water storage tank.
3. Installation techniques need to be developed to eliminate the risk of a leak in the attic space.
4. Cost of the chilled water storage tank must be reduced by approximately 50 percent in mass production to make the technology viable.

GLOSSARY

ABC	Associated Builders and Contractors
AC	Air Conditioner
ACH_{50}	Air Changes Per Hour at 50 Pascals of depressurization
AFUE	Annual Fuel Utilization Efficiency
ASHRAE	American Society of Heating Refrigeration and Air-Conditioning Engineers
BIA	Building Industry Association
Btu	British thermal unit
CFM	Cubic Feet per Minute
CFM_{50}	Cubic Feet per Minute at 50 Pascals of depressurization
COP	Coefficient of Performance
DX	Direct Expansion
EER	Energy Efficiency Ratio
EPS	Expanded Polystyrene
GTI	Gas Technology Institute
HDPE	High Density Polyethylene
HVAC	Heating, Ventilation and Air Conditioning
MHP	Measured Home Performance
MRT	Mean Radiant Temperature
PEX	Cross-linked Polystyrene
PG&E	Pacific Gas & Electric Company
PIER	Public Interest Energy Research Program
PP	Polypropylene
PVC	Polyvinyl Chloride
RTD	Resistive Temperature Device
SCE	Southern California Edison
SEER	Seasonal Energy Efficiency Ratio
SMUD	Sacramento Municipal Utility District

WCEC	Western Cooling Efficiency Center

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APPENDIX A:
SOCAL GAS RADIANT HVAC MODELING SUMMARY

The PIER project addressed in this report covers the work done in the laboratory and in the field in Northern California; cost sharing was provided by SoCal Gas to expand the analysis to similar construction located in Southern California.

In the PIER project, the energy savings is reported at 57 percent and 34 percent in heating gas energy in the two houses, and four percent to 19 percent in cooling electric energy. In this project, SoCal Gas provided match funds to conduct an analysis of the energy savings associated with the technology if installed in their service territory.

Cities were selected for the analysis based on the following criteria:

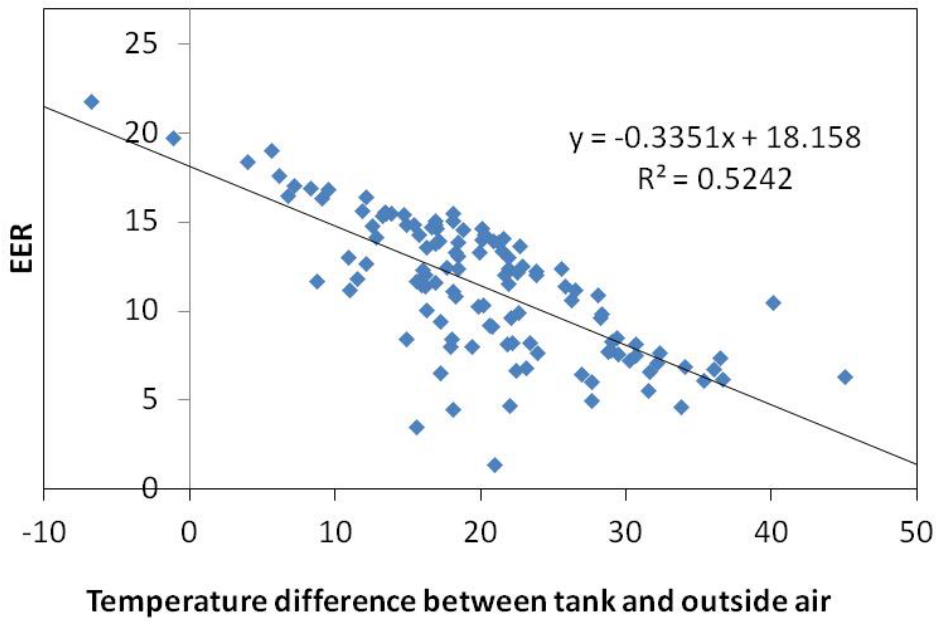
- The city must be serviced by SoCal Gas. This includes 11 southern counties.
- Larger cities (more households) were favored for a more significant impact.
- The radiant system required a supplemental electric dehumidifier in humid climates. During MHP installation, steps were taken to prevent the accumulation of excess moisture, including the addition of exhaust fans and polyethylene sheet vapor retarders under the house. Nevertheless, a dry climate is still preferred in order to avoid moisture build up on the panels that are cooled below the dew point temperature.
- BEopt used weather information from the typical meteorological year 3 (TMY3) data. In order to successfully model these houses in the chosen cities, the cities had to be near the testing site used to gather the TMY3 information.
- The analysis of the cooling utility bills suggested that the radiant cooling system worked more efficiently in cooler nighttime temperatures; this was taken into account in the analysis.
- Cities with the greatest savings potential were selected based on the BEopt simulations.

Hourly dew point values for these 34 TMY3 sites and the TMY3 site nearest to Sacramento were compared. During the cooling season, parts of the panels could reach 58°F, so the optimal climate would have dew points lower than this value a majority of the time, thus avoiding condensation on the radiant panels. Since the system worked well in Sacramento, the lowest Sacramento summer outdoor dewpoint temperature was used as a cutoff point.

Two values were calculated to compare the 34 sites with Sacramento. The first was a percentile of the total number of hours where the dew point was greater than 58 over the total number of hours. The second was the same percentile but with additional conditions: the hours must be between and including the months of May and September and the dry bulb temperature must be above 65°F (dew point percentile). These hours were more representative of the cooling hours when the radiant cooling system would have been active. The Sacramento baseline of 16 percent was used as a baseline to evaluate the other cities – above 16 percent of the hours above the 58°F dewpoint during the summer where the dry bulb is over 65°F may require supplemental dehumidificaiton.

Because the water chiller operated during the night hours for peak shifting, the operating EER value was estimated in each city by finding the average night temperature, subtracting chilled water temperature, 45°F, and then inputting the temperature difference into a linear equation of the EER efficiency at different temperatures from Figure 95.

Figure 95: Chiller Efficiency as a Function of Tank Temperature and Outdoor Temperature



Source: Western Cooling Efficiency Center

Finally, there was a calculation for the heating and cooling degree days for each location. Cooling degree days are a measure of how much the average temperature is above a cooling baseline, while heating degree days are a measure of how much the average temperature is below a heating baseline. For simplicity, the baseline of both heating and cooling is set to 65 and the amount over and under is added up for a typical meteorological year.

Table 37 shows the calculated percentiles of 15 cities, as well as the expected operating EER value of the radiant cooling system. The highlighted values are percentiles that are less than Sacramento's percentiles, or EER values that are higher, and corresponding locations are drier climates with no expected condensation on the panels. The last four are included in the BEopt testing to include more climate zones. DB is the the dry bulb temperature, while DP is the dew point temperature.

Table 37: City Dew Point Percentiles, EER Values, Nearby Cities

City	% Days Over 58°F	May-Sep DB>65 %DP>58	EER Value of Chilled Water System	Bordering Usable City

City	% Days Over 58°F	May-Sep DB>65 %DP>58	EER Value of Chilled Water System	Bordering Usable City
Sacramento Metropolitan AP	4.4	16.1	11	
Bakersfield Meadows Field	5.8	15	9	Bakersfield
China Lake Naf	.01	0.03	8	Ridgecrest
Daggett Barstow-Daggett AP	2.2	5.5	7	Barstow
Lancaster Gen Wm Fox Field	1.0	2.8	10	Lancaster
Lompoc (AWOS)	0.8	6.5	15	Lompoc
March AFB	7.6	15.1	11	Moreno Valley
Paso Robles Municipal Arpt	0.7	0.9	13	Paso Robles
Porterville (AWOS)	1.9	4.4	10	Porterville
San Luis Co Rgnl	0.5	1.5	14	San Luis Obispo
Twentynine Palms	5.3	13.2	7	Twentynine Palms
Visalia Muni (AWOS)	7.5	21.1	10	Visalia
Palm Springs Intl	10.5	24.4	5	Palm Springs
Van Nuys Airport	14.0	34.4	11	Reseda
Los Angeles Intl Arpt	27.7	72.4	12	Los Angeles

Source: Gas Technology Institute

Figure 96: Grandstaff House/Model

Source: Gas Technology Institute

Figure 96 shows one of the two houses modelled in BeOpt. The following are descriptions of the cases used to model the installation of the radiant system.

1. A "Before" case was the house prior to any modifications. This was based on input from the owners of the houses.
2. An "Insulated" or measured home performance, case was based on the initial

insulation and air sealing changes made by the installation team. Changes were made to the unfinished attic, infiltration, and mechanical ventilation.

3. A "Radiant System" case was the house once the radiant system was installed. This case differed from the "Before" and "Insulated" cases and included modifications to the Air Conditioner, Furnace, Hyrdronic Heating, Ceiling Fans, and Water Heater.
4. A "Set Point" case (Grandstaff Only) was changed from the "Radiant System" case in changes to the Heating and Cooling setpoints. These changes were reportedly made by the owner after the replacementof the system due to the fact that the system would excessively overheat and fail to cool enough at the old setpoint.
5. A "SEER 13" case was the "Radiant System" case with a more efficient air conditioner. This case was modeled since the cooling system was operating at SEER 11 in the field test and SEER13 is the National Appliance Energy Conservation Act minimum.

Each case discussed above was simulated in Sacramento and the 14 TMY3 sites chosen. The simulations were done by BEopt with information from EnergyPlus 7.2.0 and the hourly weather data from TMY3. BEopt performed a retrofit analysis over the span of a year and determined the energy expenditures for each case in a side by side comparison. To analyze this data, heating loads, cooling loads, and total loads were isolated and reported in MBtu/year.

In the BEopt simulations used in the city analysis, the average expected savings for the 6th Avenue house type were 59 percent for heating and 74 percent for cooling, while for the Grandstaff house type they were 61 percent for heating and 20 percent for cooling. These percentages were the same for almost every location, though the MBtu saved varied due to differences in usage in different climates. These savings and the other criteria were compared side by side in the full report.

The next analysis shows a cost model of the upgrades to the Grandstaff house using price assumptions from ***Radiant Heating and Cooling and Measured Home Performance for California Homes*** Section 2.2.7, BEopt simulations, and rebate values from the San Diego Gas and Electric Company.

The total cost was calculated in Table 38 for both a new construction and mature market time frame. For the mature market, it was assumed that the value of purchasing items that were not already in a mature market price would be reduced by 50 percent, an extreme value for comparison purposes.

After accounting for values of the standard equipment and utility rebates, this value was divided by the savings per year to determine the number of years until payback. The same analysis was done for each other city for comparable data. The same incremental cost-less rebate was used and divided by each respective savings per year to determine the payback in years for each location. Table 39 provides the results of that analysis.

The lifetime of the radiant system was calculated by averaging the lifetimes of the outstanding components within the radiant system, namely the air conditioner and the condensing water heater. Using lifetimes given by Appliance Magazine's *31st Annual Portrait of the U.S. Appliance Industry*, the lifetime of the radiant system is expected to be about 16 years.

Table 38: Radiant System Cost-Model in Sacramento

		New Market		Mature Market	
Grandstaff	Standard Equipment	Radiant System			
Square Feet: 1000		New Construction	Retrofit	New Construction	Retrofit
Conventional HVAC System with Ducts	$8,000				
Condensing Water Heater		$1,200	$1,200	$1,200	$1,200
Air Conditioner Condenser		$1,500	$1,500	$1,500	$1,500
Uponor Panels		$500	$500	$250	$250
Panel Installation		$583	$583	$583	$583
System Installation		$6,950	$10,425	$4,825	$7,237
Chilled Water Storage Tank		$1,750	$1,750	$875	$875
Hydronic Control System		$1,000	$1,000	$500	$500
Plumbing		$1,000	$1,000	$500	$500
TOTAL		$14,483	$17,958	$10,233	$12,646
Incremental Cost		$6,483	$9,958	$2,233	$4,646
Utility Rebate		$1,100	$1,100	$1,100	$1,100
Incremental Cost less Rebate		$5,383	$8,858	$1,133	$3,546
Savings Per Year		$113	$113	$113	$113
Payback (years)		48	78	10	31

Source: Gas Technology Institute

Table 39: Radiant System Payback in Years in Other Locations

		Payback (Years)			
		New Market		Mature Market	
Location	Savings per Year	New Construction	Retrofit	New Construction	Retrofit
Sacramento	$113	48	78	10	31
Bakersfield	$102	53	87	11	35

		Payback (Years)			
		New Market		Mature Market	
Location	Savings per Year	New Construction	Retrofit	New Construction	Retrofit
China Lake	$150	36	59	8	24
Daggett	$138	39	64	8	26
Lancaster	$135	40	66	8	26
Lompoc	$145	37	61	8	24
March	$89	60	100	13	40
Paso Robles	$113	48	78	10	31
Porterville	$110	49	81	10	32
San Luis Obispo	$102	53	87	11	35
Twenty Nine Palms	$137	39	65	8	26
Visalia	$129	42	69	9	27
Palm Springs	$126	43	70	9	28
Van Nuys	$50	108	177	23	71
Los Angeles	$35	154	253	32	101

Source: Gas Technology Institute

Five ideal locations for the radiant system were found: China Lake, Lancaster, Porterville, Bakersfield, and March. Their respective nearby cities are: Ridgecrest, Lancaster, Porterville, Bakersfield, and Moreno Valley. While Los Angeles is an attractive market opportunity, the mild climate and higher humidity levels in some areas stretch the payback period to unacceptable levels.

These locations were selected on the basis of predicted savings by BEopt modeling, the availability of their weather information, their hourly dew point averages, their estimated operating EER value, and the number of homes in the vicinity (market potential). These were the parameters given to produce cities for installation of the radiant system.

The results show that average heating therm savings of 59 percent and cooling kWh savings of 45 percent are possible based on the BEopt model, in line with the BEopt predictions for the Sacramento test sites. The payback time frame for the system is very long in Southern California's more mild climates, however, indicating that additional component cost-reduction is required.